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Compressed Air System Audit
Final Report

For

Ash Grove
Seattle, WA

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Conducted May, 2007
By
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Executive Summary

Compressed Air Consultants performed a compressed air system audit of the Ash Grove facility in Seattle, WA during the week of July 9th, 2007. The intent of the audit was to determine the existing compressed air demand along with the overall system efficiency and to make recommendations to improve the overall performance of the compressed air system while reducing operating costs.

The current state of the compressor system is good, however; significant opportunities exist to reduce the compressed air demand. Over 40% of all compressed air consumption is either through air lances being used to ensure proper product flow, or through leaks in the form of dust collector pulse jet failures and condensate drain valve failures. While the flexibility of compressed air makes it useful as a short term stop gap for issues, the overall reliance on it vs. the use of other methods of cooling and improving product flow adds significantly to the overall cost of compressed air.

Currently, each compressor and dryer is operating by their own local controls. This means that all starting/stopping, loading and unloading must occur through the compressor's onboard controls or through manual intervention. During the audit, the operating order of the air compressors was not taking advantage of the most efficient units and their respective mode of control.

Should any compressor or dryer fail, operations would be alerted by low header pressure and/or poor compressed air quality. The plant currently utilizes two distinctively different forms of compressed air dryers, refrigerant and desiccant. The refrigerant dryer is appropriate for the cement industry and the prevailing ambient conditions in Seattle, the desiccant dryer is overkill, and the plant is experiencing hidden costs associated with its operation. The refrigerant dryer has been requiring more maintenance, and during the audit, was not working correctly. The dryer was eventually repaired before the end of the week.

The proposed system recommendations will include utilizing automation that will control the operation of the system, and alert operations to situations that require service or maintenance. This will greatly reduce the non-value added time that is currently spent by the facility team, and will allow for predictive maintenance to be performed during production without affecting the system performance. The compressed air system will return to a state where the air quality is clean and dry for all operating conditions.

The estimated existing cost to operate the compressors and dryers is over $245,000 annually based upon an electrical rate of $.054/kwhr. The energy savings potential from the demand side recommendations, installing the proposed equipment, and automating the compressors is just over $80,500 per year or 33%. With an investment of $214,000, the project will yield a 32 month return on investment.

Current vs Proposed Annual Operating Costs

Constituent		kwh	Current	kwh	Proposed	Variance
Electricity (compressors)	Based on $0.054/kwh	4,478,072	**$241,816**	2,972,233	**$160,501**	**$81,315**
Electricity (dryers)	Based on $0.054/kwh	72,611	**$3,921**	85,681	**$4,627**	**-$706**
Maintenance & Labor Cost						-
Consumables Cost						-
			-			-
						-
						-
Totals		4,550,683	**$245,737**	3,057,913	**$165,127**	**$ 80,610**
	$ / yr / cfm		$87.204		$90.679	

Kwh Δ	1,492,770

Project Investment	**$ 213,929**

Simple Payback (months)	**31.8**
Internal Rate-of-Return (IRR)	**37.7%**

These savings can only be achieved with the completion of all action items

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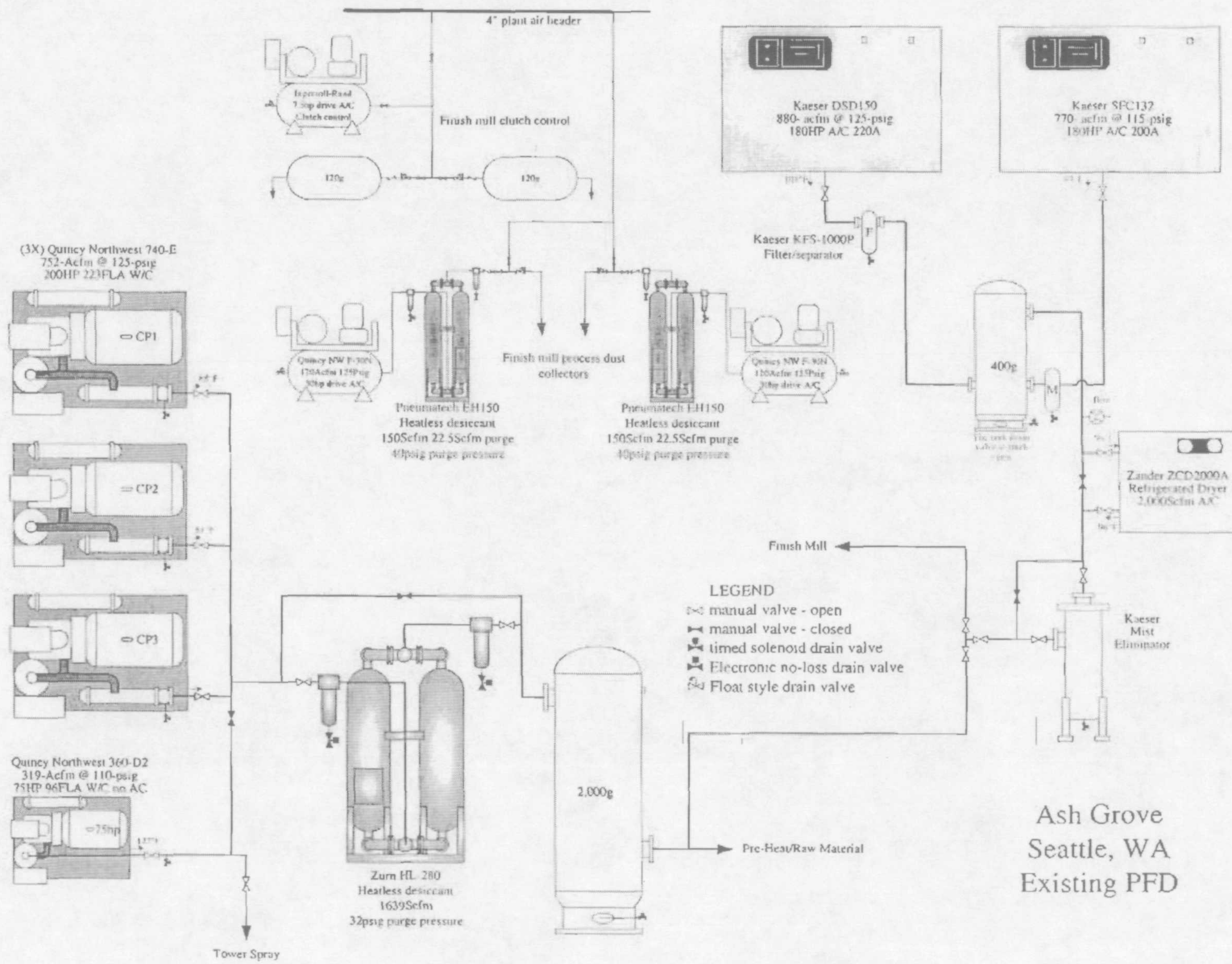
Section A
Supply



Ash Grove
Seattle, WA
Existing PFD

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Comments

Maintenance has done an excellent job maintaining and operating the compressors, however; they must spend non-value added time addressing issues within the compressed air system that should otherwise be self supported. An example of this is when a short duration demand event causes system pressure to fall; one of the back-up compressors starts. The compressor typically operates until someone from maintenance manually stops the unit. This is not due to the fact that the compressor is incapable of stopping itself, but due to the method of control. Another example is when a dryer fails, and maintenance is only alerted to this after moisture appears in the compressed air at the points of use. At this point, the entire system is contaminated and the damage has been done.

The proposed automation of the compressed air system will provide value in that it will control the compressors, and monitor temperatures, pressures, and dew-points. In the event that a component was to fail, maintenance would be alerted through the existing DCS, and immediate action can be taken. Should the communication link between the DCS and the compressors be broken, the compressors will revert to their local on board controls and respond to demand changes.

Compressor Rooms

There are three areas where compressors pump air into the plant air system. Beneath the burner building are two Kaeser compressors that pump through filtration and a refrigerant dryer and into the common plant header. Beneath the pre-heat tower are three Quincy Northwest compressors also dedicated to the plant air system that discharge into a common header before flowing through filtration and a desiccant dryer. A fourth Quincy Northwest compressor is in the room but dedicated to the tower spray system. The third room is in the finish mill where two Quincy Northwest compressors followed by filtration and desiccant dryers that are dedicated to each mill's process dust collector. A 7.5hp compressor exists in the finish mill to supply air to the clutches in the event that plant pressure falls below an acceptable pressure. This compressor is normally off and was off during the audit.

Issues

There are no issues with having multiple compressor rooms in a system, however; when they are isolated like the finish mills and the spray system, the compressors typically are left to operate partially loaded (inefficient), and in the event that one were to fail, the pressure would fall to below critical levels before any one could respond to open the isolation valve to allow plant air to support the demand.

A majority of the time, the spray system operates just to support the leaks within the piping of the spray system until such time as the spray system needs compressed air. This may not seem significant, but the power consumed to support the whisper of leaks is over 50hp or $19k annually.

The finish mill compressors are in a dusty environment and are operating independently to support the needs of their respective process dust collectors, leaks, and the purge air requirements of their desiccant dryers. Each compressor is operating in modulation at 70% load but pulling 90% of the power.

Actions

This plant can and should be operated as one common system. The pressure requirements for the process dust collectors in the finish mill are not higher than the plant compressors are operating at, so the air for the pulse jets could come from the plant header. The valves that are closed to isolate the 30hp finish mill compressors from the plant should be opened and the 30hp compressors along with their desiccant dryers should be turned off.

The spray system air compressor is only required to supply air to the nozzles less than 50% of the time. Recommendations will be made to install a reverse acting regulator to prioritize the spray system pressure, and spill any excess air from the spray system to the plant air system. This will fully load the 75hp at all times, either directing all the air to the plant or to the nozzles when necessary. The compressor is not equipped with an aftercooler or moisture separator/drain, and it will be recommended that these be installed along with water piping. A flow check valve should also be installed so that in the event that the 75hp compressor was to fail, its air could still be delivered from the plant air system.

Burner Building

The Kaeser compressors are identical in external appearance but different in their displacement and control. The units are noted as a DSD-150 and an SFC-132. The DSD-150 is rated for 883Acfm at 100psig. It is only capable of being operated in online/offline, and its offline (unloaded) power is 34kW. The total package power at the rated flow and pressure is 143.5kW, meaning the specific power for the unit is 16.3kW/100Acfm. This is the most efficient fully loaded compressor on site. The SFC-132 is rated for 810Acfm at 100psig. It is equipped with a variable frequency drive that controls the speed of the motor in an attempt to match the output with compressed air demand. The total package power at the rated flow and pressure is 139.9kW, and the specific power is 17.1kW/100Acfm. This is the most efficient partially loaded compressor on site. Neither Kaeser compressor is equipped with an integral moisture separator.

The DSD-150 discharges as a 2" line into a Kaeser KFS-1000P 1000Scfm combination filter/separator that has a Kaeser electronic condensate drain valve. The SFC-132 discharges as a 2" line into a custom built separator that has a Kaeser electronic condensate drain valve. The air flows from each separator into a 400-gallon receiver tank that also has a Kaeser electronic drain valve. From the receiver tank, the piping expands to 4" flowing into a Zander 2,000Scfm non-cycling refrigerant dryer and then through a Kaeser mist eliminator filter before entering the header. There is an insertion style flow meter installed in the 4" line at the outlet of the 400-gallon receiver tank. The dryer and mist eliminator filter are equipped with timed solenoid condensate drain valves.

Comments

Ideally, because of the ability of the SFC-132 to efficiently speed up and slow down, the dryer and filter will see varying flows. The existing dryer does not have the ability to do this, so its refrigerant compressor must constantly run. The dryer does have the ability to start and stop its condenser fans and can also throttle a hot gas bypass valve to prevent freeze ups during low load conditions. When the potential exists for sustained flows below the dryer's full capacity at rated conditions, the most efficient choice of dryer would be one that had the ability to start and stop, or utilize a frequency drive. This would ensure the minimum amount of energy consumption during periods of partial loads. It does not make financial sense to replace the dryer, however; should the dryer become too maintenance intensive or catastrophically fail, then it is recommended that a cycling refrigerant dryer be considered.

The location of the mist eliminator filter is downstream of the refrigerant dryer. It has been our experience that the best location of the mist eliminator is to be upstream of the refrigerant dryer, thereby protecting the dryer's evaporator from any oil carried over from the air compressor. Should the dryer need to be replaced, then it is recommended that the filter be re-piped so that it is upstream of the dryer.

Current Issues

The refrigerant dryer was not working during the first three days of the audit but was repaired on day 4. The drain valve on the 400-gallon receiver tank has failed open and is exhausting more than 50Scfm of compressed air. The discharge temperature of the DSD-150 was 102°F. The flow meter is installed in the untreated compressed air piping and is reading more air flow than the compressors can possibly make.

Actions

The drain valve on the receiver tank should be repaired or replaced if necessary. The upstream separators are equipped with traps, and these should be inspected periodically. The discharge temperature of the DSD-150 was much higher than the SFC-132 which is due to the load. The design of the compressor's internal cooling fan and aftercooler is to be able to cool the outgoing compressed air to within 20°F of the compressor intake air when the ambient relative humidity is 40% or less. The ambient conditions during the readings were 78°F at 40%RH, indicating that the cooler is becoming fouled and in need of cleaning. Each of the Kaeser units are air-cooled, so the package discharge temperatures should also be monitored.

The insertion style flow meter is sensitive to moisture. It is currently installed in the 4" piping upstream of the dryer and filter, so the air that it is monitoring is normally 100% saturated. When possible, it is recommended that the flow meter be re-installed downstream of the clean-up equipment.

The proposed automation will include compressed air dewpoint monitors to be able to monitor the moisture levels in the air at the outlet of each compressor room. If the dewpoint begins to rise, the automation will signal an operator to alert the maintenance team to a dryer or drain trap failure.

Pre-Heat tower compressor room

There are four air compressors in this room, three identical Quincy Northwest 200hp units and one Quincy Northwest 75hp unit. The 200hp compressors are tied to the plant air system and the 75hp unit is dedicated to the tower water spray system. The 200hp units are noted as a 740-E and are rated for 752Acfm at 125 sig. Each is only capable of being operated in full range modulation or in online/offline. The total package power at the rated flow and pressure is 135kW, meaning the specific power for the unit is 17.96kW/100Acfm. Each of the three compressors can be operated in local mode as continuous run or through an external sequencer. Currently, each unit is being operated locally. The 75hp compressor is a 360-D2 rated for 319Acfm at 110 psig. The total package power at rated flow and pressure is 58kW for a specific power of 18.24kW/100Acfm. All four of the units are water-cooled designs, with the exception being the 75 hp is not equipped with an after-cooler or a moisture separator.

The three 200hp units discharge into a common header before flowing into a Zurn HL-280 heatless desiccant dryer rated for 1639Scfm at 100°F and 100 psig. The dryer is preceded by a coalescing filter and followed by a particulate filter. The dryer pre-filter and compressor discharge moisture separators are equipped with timed solenoid condensate drain valves. A 2" line connects the common header of the 200hp units to a tee at the outlet of the 75hp. A service ball valve in the line is closed keeping the two systems separate.

Comments
The plant air system is using two distinctly different forms of air dryers. Each is in place to accomplish the ultimate mission of ensuring the outgoing compressed air is free of liquid water for all operating conditions, however; the main difference is the ultimate cost of operation. The Zander refrigerant dryer is consuming approximately 10hp to dry up to 2000Scfm which equates to nearly $4000/ year. The restriction to air flow at 2000Scfm at 100psig inlet is 3psig. The pressure drop forces to compressors to work 1.5% harder to meet a header pressure of 97psig. The dryer can reduce the compressed air pressure dewpoint to 35-40°F which is excellent for the conditions at the plant. The Zurn desiccant dryer consumes very little electricity, but does need to use compressed air to regenerate a saturated tower. The dryer needs 15% of its rated capacity (246Scfm) to purge in order to maintain a -40°F air pressure dewpoint at the outlet. The cost to generate 246Scfm of compressed air is over $21,500 annually. The restriction to air flow across the dryer and filters is between 8-10 psig when the filters are new and will rise to 16 psig before the filters are ready to be changed. This forces the compressors to work 4-8% harder meaning more kW consumed than necessary to meet the plant air quality required.

Issues
The 75hp compressor is segregated from the main system to support the needs of the spray system only. When the spray system needs the compressed air (less than 50% of the time) it is operating at full flow, the remainder of the time it is operating at 70% of full power to generate 0-1% flow. The times that the compressor is not pumping air into the spray system, it could be redirecting the air to the plant system.

Actions
It is recommended that a 2,000Scfm mist eliminator filter and 2,000Scfm cycling refrigerated air dryer be installed in parallel with the existing desiccant dryer. The intent is to allow the compressed air flow to be treated by the refrigerant dryer during the periods of time when the ambient temperatures do not fall below 40°F. When the ambient temperature does fall below 40°F, the desiccant dryer should be energized to allow the -40°F air to blend into the header with the +40°F refrigerant dried air from the burner building compressor room. This will eliminate the need to purge 246Scfm of air a majority of the time.

It is recommended that a shell in tube heat exchanger (after-cooler) and moisture separator be mounted to the 75hp compressor. This will allow the compressor to pump air into the common discharge header with the 200hp Quincy machines. It is also recommended that a 1.5" reverse acting regulator be installed to maintain a constant pressure in the spray tower header, and to prioritize the 75hp compressor to the spray system when it is operating. The regulator should be set to a higher pressure than the discharge pressure of the 200hp compressors. It is also recommended that a single 2" pipe with a flow check valve be installed from the dry header so that in the event that the 75hp compressor was to fail, the spray system could be fed from plant air. The purpose of taking the air from the dry header is to ensure that the air will get to the spray system even if the 200hp Quincy compressors are off.

It is recommended that the timed solenoid drain valves be replaced with pneumatic no air loss drain valves.

Compressor Mode of Control

To understand how the potential for horsepower reduction exists through proper compressor control and sequencing, it must be understood what modes of control exist on the compressors at Ash Grove.

Online/Offline

The Kaeser DSD150 is only capable of online/offline control but the Quincy Northwest compressors can be operated in this mode of control or in modulation. The Kaeser compressor senses pressure at its discharge, and this signal is monitored by a pressure transducer. The Quincy Northwest units monitor pressure at the minimum pressure check valve and send the signal to a pressure switch. The pressure switch is designed to have a maximum pressure on one setting, and then a second adjustment known as span to set a minimum pressure. The Kaeser units utilize a pressure transducer and a microprocessor. The pressure transducer sends an electronic signal (0-5vdc or 4-20mA) to the microprocessor which operates between an operator adjusted load and unload setting. When the pressure at the sensing point falls to the minimum pressure setting, the inlet valve of the compressor opens, allowing air to be ingested into the compressor. The compressor then begins to pump all the air it can until the pressure at the sensing point rises to the unload setting. At this point, the inlet valve closes. The compressors will then open a valve to unload the pressure within the sump to a level necessary to maintain an adequate lubricant pressure. There still is some air intake to the compressor that is required to prevent the rotors from cavitating. It must be noted, that for the Quincy Northwest machines to be operated in online/offline, the subtractive pilot valve must be adjusted closed so that no air pressure at the minimum pressure check valve can force the inlet valve to modulate.

When operating in online/offline, the compressor will be operating at full power when loaded and the discharge pressure is at the package rating. The rotary screw compressor will be operating at minimum power when the compressor is unloaded and the sump has fully depressurized. The variations in the performance curve shape are due to the time it takes for a compressor to blow down the sump pressure. The graphs in the attachment section illustrate the theoretical performance versus the realistic performance. The factors that shape the realistic performance curve are the displacement of the compressor, the designed minimum sump pressure, the time to blow down the sump, the pressure band that the compressor operates within, and the available volume that the compressor pumps into.

The positives to operating in online/offline are that it is the most efficient form of control next to frequency drive when the above noted factors are ideal. The negatives to this form of control are that the more frequently the compressor loads and unloads, foaming of oil within the sump will increase, carry over of oil through the lubricant separator will increase, and the stresses applied on the airend bearings.

Modulation
The Quincy Northwest compressors have the ability to adjust the position of the compressors inlet valve to something other than fully open or fully closed. When operating in modulation, the compressor senses the discharge pressure, and as it begins to rise above the load pressure, the inlet valve will begin to close in an effort to match compressor output with demand. The compressor senses the pressure at the minimum pressure check valve on the dry side of the sump separator element. It then regulates the signal through a subtractive pilot valve to position the inlet valve. The adjustment of the subtractive pilot valve determines the pressure at which the inlet valve begins to modulate. Depending upon the setting of the subtractive pilot valve, it is possible that these units can modulate from 100% to 0% of full flow.

The positives to operating in this mode of control are that the compressor can maintain a constant discharge pressure if the subtractive pilot valve is adjusted correctly, and if demand events do not occur faster than the compressor can respond. The ability to throttle the inlet valve means that the compressor will not load and unload which means less foaming of the oil, and less fatigue on the bearings.

The negatives to operating in this mode of control are that the compressor will draw 80% of its full power to pump 0% of its air. The specific power curve for a compressor in modulation is the least efficient. When a compressor operates in extreme modulation for extended periods of time, condensation can begin to form in the sump which could damage the bearings and the minimum pressure check valve.

Variable Frequency Drive (VFD)
The Kaeser SFC-132 compressor is equipped with a variable frequency drive (VFD) to control the speed of the main motor. As the pressure begins to rise at the package discharge, a pressure transducer sends a signal to the microprocessor which then tells the frequency drive to slow the motor speed down, which also slows the rotor tip speed. The compressor displacement is directly proportional to the rotor tip speed. The Kaeser unit is designed to turn down from 100% to 25% of its full displacement. This is the most efficient compressor when operating at less than 100% load.

Current method of control

During the audit, the normal mode of control was to have the DSD-150 fully loaded, followed by one or two of the Quincy Northwest units to be manually started and stopped as necessary. The SFC-132 was acting as the trim compressor while the demand was at its peak. On the third day of the audit, a failed pulse jet on a dust collector had been repaired, and demand fell causing the SFC-132 to unload and turn off. Because the units are in local control, the next machine to trim was the DSD-150. It began to load and unload, and also one of the Quincy Northwest began to modulate its inlet valve. Normally, maintenance is alerted to this during production meetings in the morning, and they will manually turn off one of the Quincy Northwest units.

The SFC-132 should always be the first compressor on and the last compressor off, and always be the trim compressor. In local settings, this could be achieved by adjusting the unload pressure to be the highest of all units and the load pressure to be the lowest of all units to be operated. The maximum operating pressure of the SFC-132 is 110psig, therefore, all other compressors will have to have their unload pressures set below that number. The difficult part of adjusting the local set points is to also ensure that this compressor is the only machine that can partially load. In this case, the unload settings of the subsequent machines would have to all be above the target pressure setting of the SFC-132. This would be incredibly difficult to manage based upon the dynamic nature of the compressed air demand at the plant. This is why it is recommended that automation be incorporated.

After-cooler/Moisture Separator/Drain valve function

The function of the compressor's after cooler moisture separator and drain is critical, as over 50% of all moisture ingested by the compressor should be removed by the drain at the discharge moisture separator. The remaining 50% of moisture vapor should be dealt with by the filters and dryers. At 80°F and 50%RH, the 200hp compressor will ingest 5 gallons of water vapor per hour when 100% loaded. If the after-cooler, moisture separator, and drain valve are functioning as designed, there would be just more than 2.5 gallons per hour of condensate being pumped downstream to the filters and dryer.

The compressor after-cooler is designed to remove the heat from the compressed air that was generated as a result of compression. The manufacturer typically designs the after-cooler to reduce the outgoing compressed air temperature to within 20°F of the cooling medium temperature, air or water. The performance of the cooler depends upon multiple factors, some of which include the ambient relative humidity (air), and volume of water flow in gpm. The ability of the cooler to exchange heat with the compressed are has an impact on dryer performance, as for every 5° F of temperature rise. The moisture holding capability of the compressed air goes up by 25%.

The purpose of the separator is to allow the moisture that was condensed in the after-cooler to be removed from the air stream, at which point, it is the condensate drains job to evacuate the liquid. The Zander dryer and the mist eliminator filter as well as the drain points in the pre-heat tower compressor room are using timed solenoid drain valves to perform this task. A timed solenoid valve has two sets of control, time open and time closed. The amount of moisture that must be removed from the system is dependent on the ambient air conditions at the inlet to the compressors, and these conditions are constantly changing. If the operator does not change the settings that the timed solenoid valve is open and the time the valve is closed based on the changing conditions, compressed air will be wasted or condensate left in the system. In the warmer months, the amount of moisture in the system can be sixteen times that of cooler months. Timed solenoid drains are inexpensive and widely used throughout the industry, but for reasons mentioned above, are not conducive to long term energy savings.

It is recommend that zero air loss pneumatic drains with condensate level switches be installed to replace each timed solenoid and float drains. The minimum port size is 3/8" eliminating any plugging problems from rust and scale from the carbon steel piping. The drain capacity is 16-gpm and the condensate reservoir is large, reducing the frequency of cycling for heavy condensate flows. These drains are very reliable; as the most sophisticated clean up system must have a reliable drain to function properly. Because of the potential for freezing, it is also recommended that each drain valve have a heating element installed. Drains are not maintenance free and must be inspected on a regular basis.

Automation

Currently, each supply side component is controlled locally. Should there be a sudden change in demand, or perhaps a failure of a component, then something must happen on the demand side to alert the maintenance team to respond. If the pressure were to fall to unacceptable levels due to the failure of a compressor, critical applications will protect themselves by shutting down. The penalty is the delay in operation due to the inconsistency of the compressed air. In the proposed arrangement, the compressor's local controls will be able to respond to changes in system pressure without operator intervention, however, we feel that an elevated form of automation is required to ensure the compressors are always operating in the most efficient manner.

This elevated form of automation should

1. Always ensure that the SFC-132 is the trim machine
2. Sequence the Quincy and Kaeser DSD -150 machines to balance hours for maintenance purposes.
3. Detect and react to compressor failures by monitoring amps.
4. Electronically control multiple machines in smaller deadbands and use rate of change logic to determine the best use of the variety of horsepower.
5. Monitor and benchmark the ever-changing operating conditions (pressures, temperatures, and pressure dewpoint).
6. Monitor the pressure dewpoint of compressor station to determine dryer failures.
7. Determine if a pneumatic no-air loss drain valve has failed by monitoring float level switches.
8. Monitor pressure in dust collector pulse manifolds to determine when a pulsed jet fails open, and to identify what dust collector has the issue.

The plants operating automation platform is Allen-Bradley PLC5. Any future compressor automation should be able to seamlessly communicate with the existing platform.

We believe that Ash-Grove would benefit from this elevated form of automation by allowing the software to determine the most energy efficient form of operation, by immediately alerting the facility to issues with the compressed air system, and by freeing up the non value added time maintenance currently spends working with the compressed air system.

Storage

The amount of storage in the volume of receiver tanks and piping in the system is low measuring just under 4,800-gallons. Within the dry header piping, there is just one 2,000-gallon receiver tank. All other tanks and manifolds are less than 120-gallons each. Storage in any compressed air system represents time. The more time the compressors have to respond to changes in system pressure, the more accurately the system pressure can be controlled. This also represents time for the system to recover to short duration demand events without the need to start air compressors. In the event that the largest compressor operating was to fail (880Acfm Kaeser DSD-150), the system pressure could fall as fast as .33 psig/second. This would only give operations 30 seconds to respond to start a backup Quincy before pressure would fall 10psig. Other events of large magnitude such as unregulated open blowing through air lances at the raw material transfer towers, or even a failed dust collector pulse jet could cause the same type of system reaction.

Storage can be sized according to how large and long a demand event is, or by determining how low the system pressure can fall in the event that the largest compressor were to fail before the equal amount of backup compressor horsepower can be started. When controlling a system through automation without demand control, this system will have to rely upon the ability of the SFC-132 to determine how the system pressure fluctuates. The automation will determine what speed the frequency drive is operating at, and determine what compressor needs to be started or stopped to most efficiently match demand. When the 200hp units are started and stopped, the rate of pressure change in a small volume system such as this one will be greater than the frequency drive compressor's ability to maintain a stable pressure.

The calculation below was made to determine how much storage would need to be added to the system to give the SFC-132 time to accurately respond, and in the event that the SFC-132 were to fail, limit the amount of cycles the largest trim compressor would encounter when operating at a worst case condition of being 50% loaded.

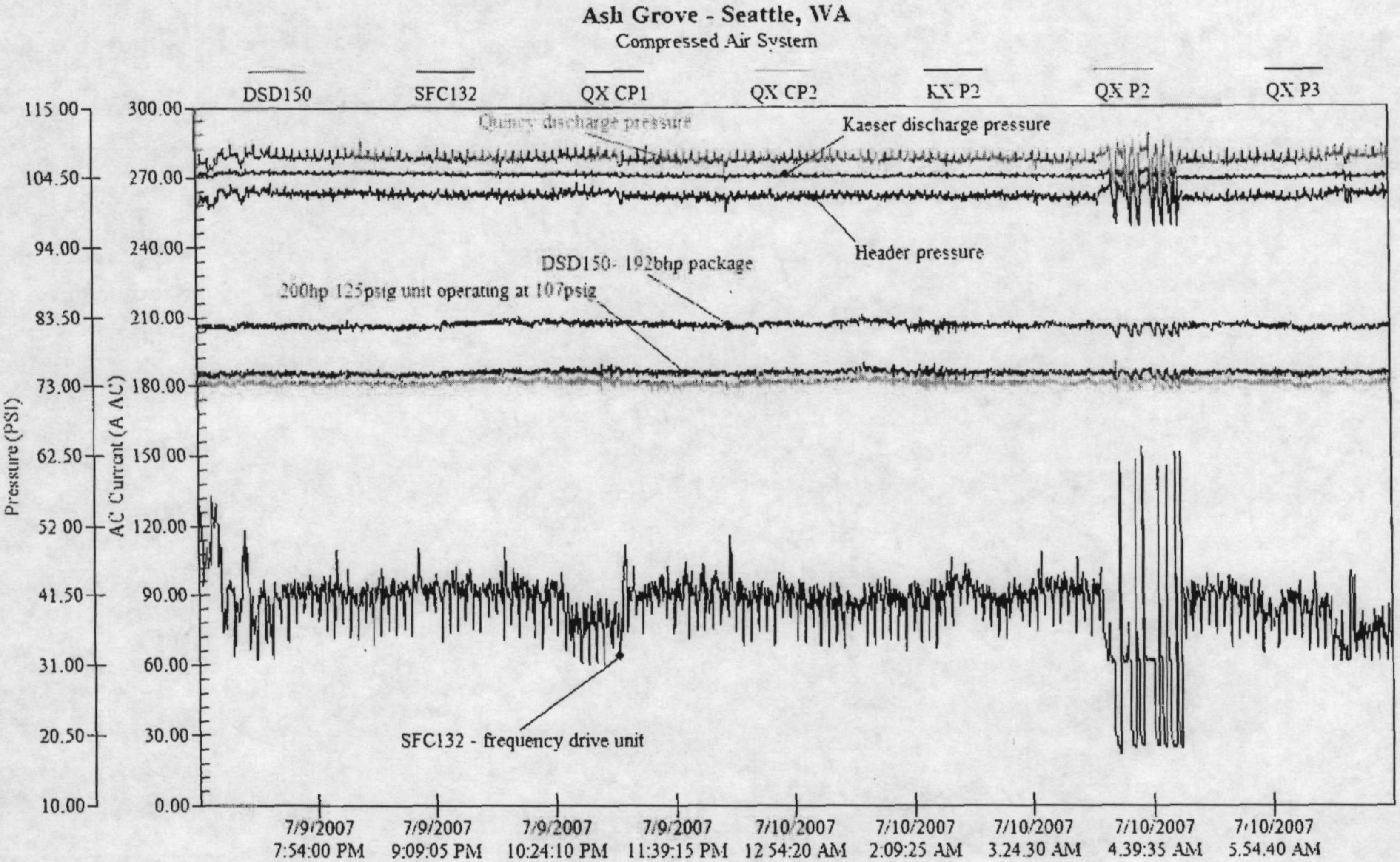
Actions

The existing volume of 4,800-gallons of storage is not sufficient, and needs to be increased for accurate system control. It is recommended that a minimum of 6,000-gallons of storage be added to the dry side of the system. The tank only needs to be teed into the existing header with a service valve.

Master Signal

We are recommending the installation of a master signal line from the dry header or receiver to each of the compressors. By taking the signal from downstream of the cleanup equipment, the compressors will respond based upon a system pressure versus a pressure zone that constantly changes due to varying differential pressures of a filter element and a dryer. This means that when in local control, each air compressor will be responding to the same pressure, unlike today, when each compressor is responding to an air signal from within the sump vessel which does not account for the pressure losses through the minimum pressure check valve, after-cooler, filters, and dryer. Each arrangement will have its own pilot unloader valve to protect the compressors in the event that a block was to occur between the master signal and the compressor. This valve would sense the higher pressure immediately downstream of the compressor, lift the valve and override the master signal, allowing the compressor's internal transducer to sense this pressure and unload the machine if necessary.

Section B
Data



The graph represents the compressor amp and system pressure profile. This illustrates the compressed air demand requires that two of the three Quincy Northwest units as well as the Kaeser DSD150 are operating at full load and flow while the frequency drive Kaeser SFC-132 is modulating to meet the variance in demand. The anomaly at 4:39am on the 10th is when the trim demand falls below 25% of the output of the SFC-132, at which point the compressor begins to load and unload.

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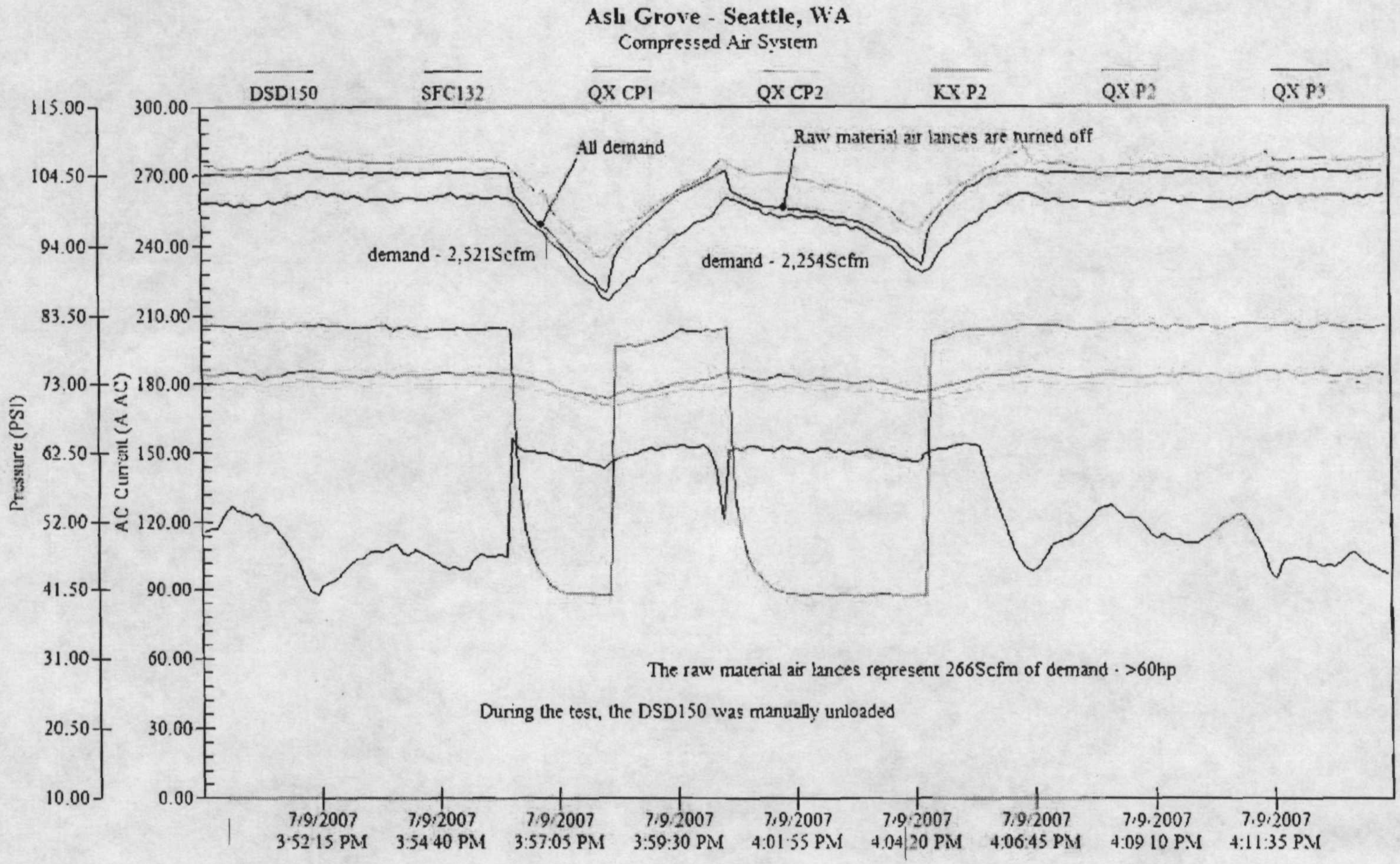

The purpose of this graph is to illustrate a test of measuring the size of a compressed air demand user. The first test was performed when two air lances were on in the raw material chute and the DSD150 was manually unloaded. Notice the power spike of the SFC-132 attempting to pump more air to prevent the pressure from falling. At this point, the SFC-132 is fully loaded and the pressure falls at a certain rate. During the second test, both air lances were turned off and the DSD150 was again unloaded. At this point, the rate of pressure change (fall) was very gradual, indicating that the plant could nearly operate on three compressors if the air lances were not being used.

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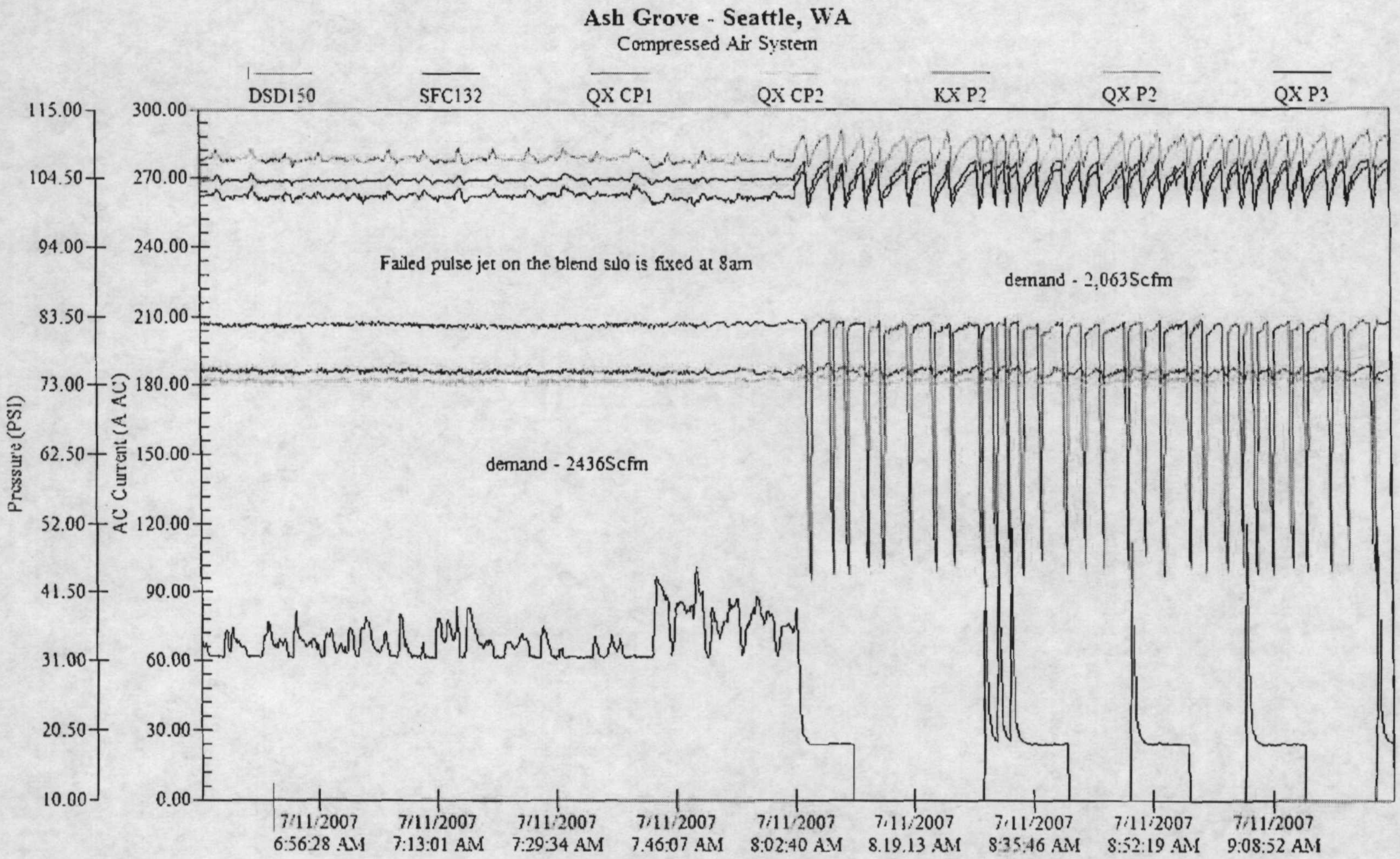

This graph illustrates the effect on supply when a failed dust collector pulse jet was repaired. When the air demand disappeared, the SFC-132 and DSD150 compressors responded by first unloading, followed by the SFC-132 turning off. The DSD150 then becomes the trim machine, and the SFC-132 starts periodically due to settings of start timers and low system capacitance. This is a great example of why automation is necessary to ensure the most efficient (operator free) operation of the compressors for all conditions.

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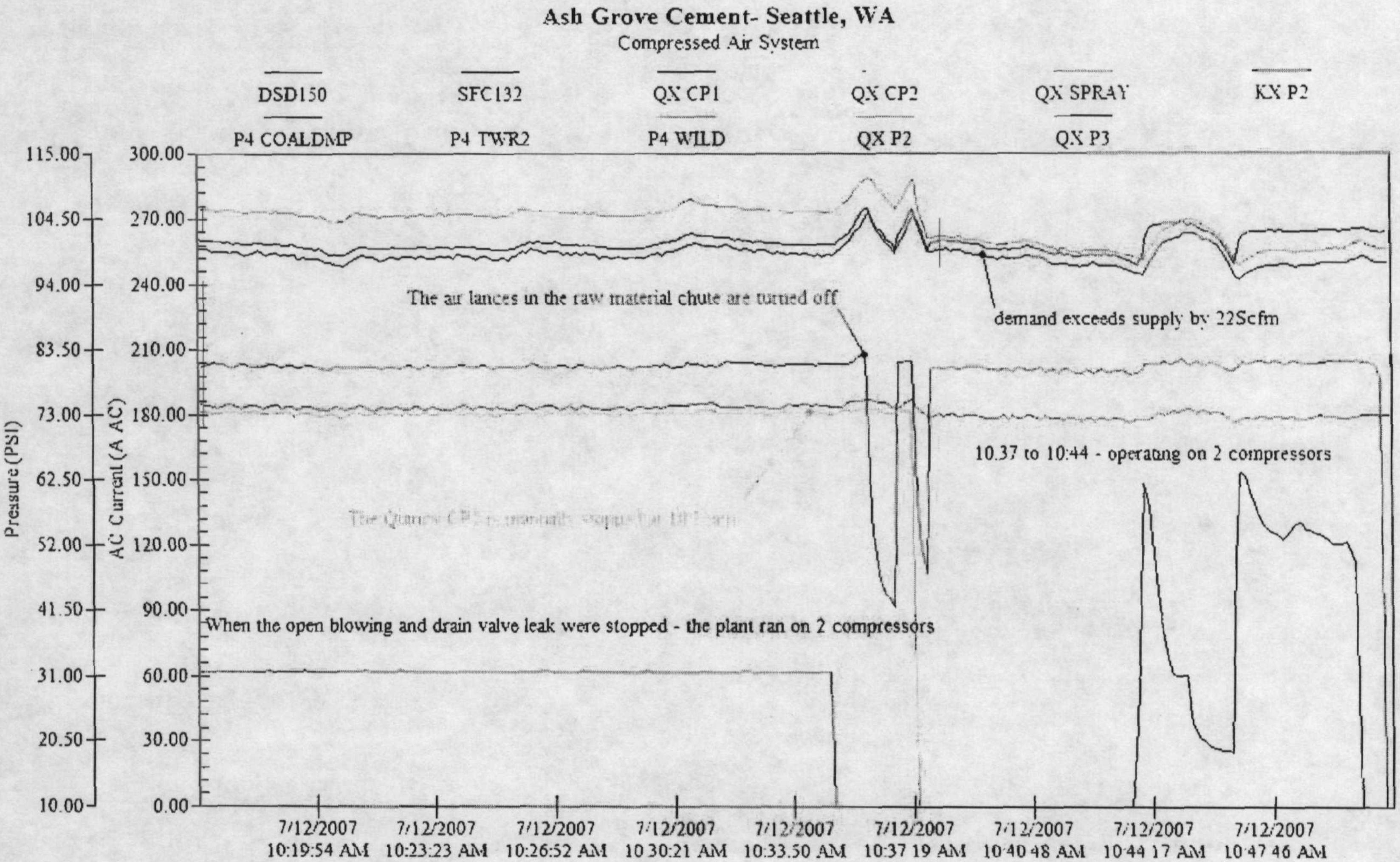

On the final day of the audit, a test was performed to determine the effect on supply when the demand consumers of the air lances, bearing cooling, and failed drain valve were all stopped. As can be seen from 10:37 to 10:44am, the only two compressors were operating and the deficit was only 22Scfm!. The plant can and should expect to operate with this goal in mind.

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Section C
Demand

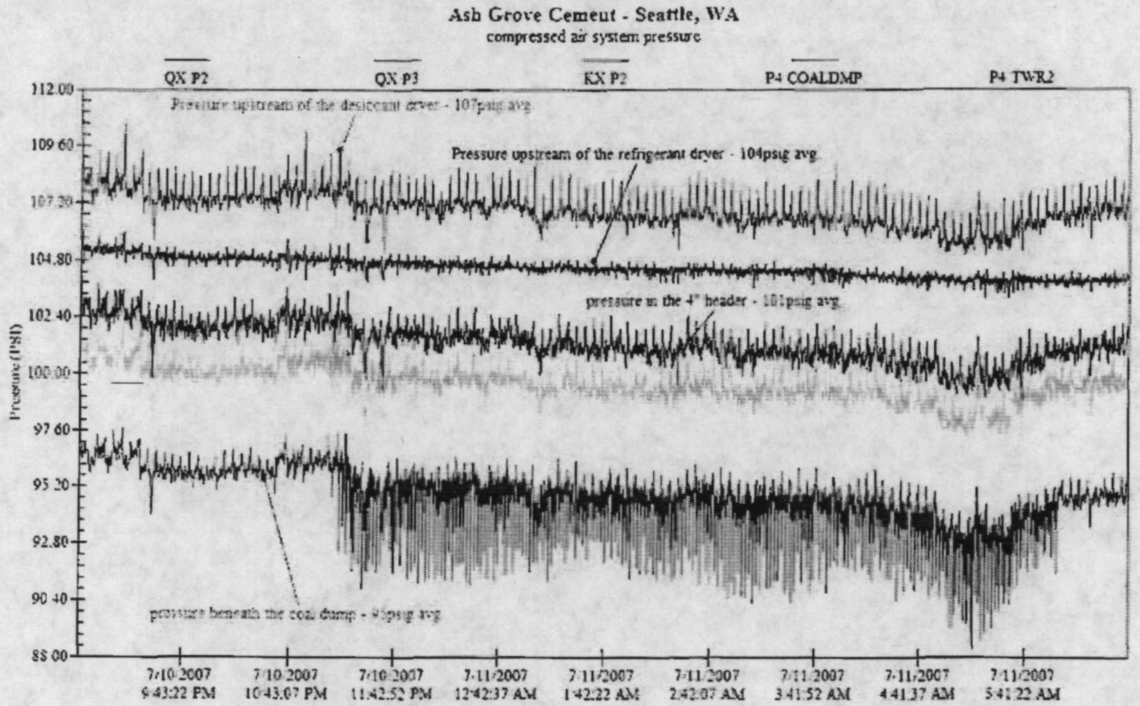
Piping network

The piping network in the facility is excellent for supplying air to the critical users, but sub-par for supplying air to the remote areas such as the barge unloading and coal dump areas. The supply of compressed air is adequate for every user in its current arrangement, but there are some changes that would enhance the air supply to all sections of the plant.

The main header piping is 4" that leaves each compressor room and flows to the coal mill to the north, to tower #11 to the west, to the clinker silos to the south, and to the finish mill to the west. The 4" piping is connected from north to south by a 2" pipe that travels beneath the clinker coolers.

Issues

The following illustration is to show the difference in pressure at various locations in the plant when measured simultaneously. The lowest measured pressure was the area beneath the coal dump. Inspection of the piping showed that this was the point furthest from the supply compressors when considering pipe length. The pressure in the coal dump area is averaging over 6psig less than the main header pressure.



Comments
There is no issue with the supply of compressed air to any section of the plant. The average header pressure of 101psig is over 20psig higher than the highest required article pressure at the point of use, so the opportunity exists to reduce compressor energy by reducing their operating pressure. If the plant were to operate at an average of 85psig, the compressors could be operated at 15psig lower pressure, and consume 7% less horsepower. When operating at 85psig, the difference in pressure in the coal dump area could be greater than 6psig. The action plan demand changess for the plant are to remove the compressed air lances from the raw material chutes that are using air from the piping immediately upstream of the coal dump. When the demand of air is removed. the pressure drop from the header to the coal dump will be greatly reduced.

Actions
A single 2" piping connection could be made from the clinker storage building 2" pipe to the piping that feeds the coal dump. This will eliminate the pressure drop that results from the piping being a branch at the end of the network.

Demand Overview

The demand in the plant during the time of the study was roughly 2800 Scfm and is broken out as follows:

Application	Current	%	Proposed	Aggressive
POU Desiccant Dryers and Auto Drains	323	11%	100	100
Dust Collectors	583	21%	528	378
Open Blowing, Aeration	574	20%	227	181
FK	157	6%	140	140
Cameras, Peepers and Cab Coolers	112	4%	102	102
Gear Spray, Oil Atom Flame Control	8	0%	8	8
Water Spray and Grinding Agent	20	1%	20	20
Downcomer	273	10%	273	273
Vibrators	0	0%	10	10
Air Cannons	11	0%	9	9
Miscellaneous, Undiscovered leaks	214	8%	200	200
Leaks inc DC	543	19%	200	100
Total	2818		1817	1521

It is estimated that the potential reduction in demand is between 1000 and 1300 Scfm. This opportunity exists due to the extensive use of compressed air for open blowing, the discovery of a failed solenoid valve on one of the dust collectors and some other opportunities with regards to the operation of dust collectors, FK pumps and the repair of leaks.

The primary concerns in the projection of this reduction are two fold. First, how representative was the operation of the dust collectors and open blowing/air lances during the time of the study. The vast majority of dust collectors were pulse on demand and roughly half of all dust collectors were not pulsing. The unanswered question is how great the variation in pulsing for the dust collectors is.

The second issue is how representative this was with regards to the use of air for bearing cooling, air lances and the like. Given the variation in weather, this could be significant.

When it comes to achieving these demand reductions, the primary concern is not one of technology but one of willpower. These savings can be achieved without impacting production; however, they will require a change of behavior on the part of the plant. If air is used judiciously and discriminately, this can be achieved with minimal effort, but it does require a change in mindset which is often quite difficult. Therefore, a vital component to this particular project is training of operator personnel of the cost of air as well as ongoing monitoring of system operation.

One final note, once this project is implemented; there will still be opportunities to further reduce demand by fine tuning some of the smaller users in the system. The purpose of this first phase of the project is to get the system under control and creating confidence in the system's operation.

Desiccant and Membrane Dryers

There are several desiccant dryers in the system including point of use dryers which use compressed air as part of their process. In addition to the desiccant dryer serving the Quincy compressors, there was a small unit in the Quincy room on the wall and two units in the finish mill, one of which was on line. There was also membrane drying for the HMD20-6 particle analyzer as well as a membrane system for the Prism nitrogen generator.

Points of use dryers are useful if the supply air is untreated or the main dryer fails; however, there is significant cost associated with this application. Considering that in a system where the air is already dried, they do very little for the improvement of air quality. Most of these dryers use 15% of their capacity as purge air regardless whether the application is running or if the air is already been dried. There is also evidence that if the air is already dry, the dryers cannot work properly as they are dependent on heat generated by the desiccant bed when the water molecule is adsorbed by the desiccant. This exothermic reaction stores heat in the desiccant to be used in the de-sorption cycle. If the moisture is significantly low, then insufficient heat will be generated for the dryer to dry properly. If the air is dry already, then there is no "wetness" penalty, however, the energy is still wasted.

Once air is exclusively supplied from the Kaeser system, the two desiccant dryers in the finish mill, the main desiccant dryer and the small unit in the Quincy room can be bypassed and taken off line.

Given the importance of the drying for the particle analyzer and the minimal 4 cfm it consumes, taking this unit off line is not recommended. We are assuming that the nitrogen system will continue to use air as it currently does.

Dust Collectors

The extensive use of pulse on demand controls minimizes excessive pulsation keeping demand down. The three opportunities on the dust collector side are reducing the pulse duration on some of the critical dust collectors, reducing the supply pressure, and fixing a blown pulsejet. The demand during the study was 580 Scfm (significant figures) and assuming the sample was representative, this demand could be reduced to at 230 Scfm and possibly even 180 Scfm.

Before we get into our specific findings, it is often helpful to review the basics of the bag houses and dust collectors. Compressed air is the energy needed to clean the bag by rearrangement of the dust cake on the pulse. The design intent of the pulsejet is to create a short duration of compressed air release that shocks the atmosphere in the blowpipe that generates a pressure pulse directed into the cage and bag. The result is a pressure bubble that traverses the length on the bag similar to dropping a soft ball into the bag. The ball flexes the bag causing it to expand and breaking the cake structure that forms on the bag. The pressure bubble as it is introduced into the cage or venturi causes a negative to occur at the top of the bag in the clean air plenum which pulls air from the plenum and that air follows the bubble down the bag. This air will exhaust through the fabric from the clean side to dirty side and will move the dust that has been fractured by the pressure bubble down the bag as well as rearrange the dust cake resulting in a change in differential or resistance to air flow.

Due to the rearrangement of the dust during cleaning and the proximity of the rows of bags it is not recommended to clean any pulsejet sequentially. Reason being when one row cleans and sees a reduction in differential pressure it can receive more air and pulsing the row beside it will produce suspension of fines that will migrate to the clean row. This can cause the separation of fines from the dust cake and over a period of time produce a dust cake of similar particle size that cannot be rearranged resulting in no differential change on pulsing. Pulsing should be staggered and the recommended sequence would be 1,4,7,10,etc until the last valve on the manifold pulses then pulse 2,5,8,11,etc until the end and then pulse 3,6,9,12,etc, always pulsing a row between dirty rows. This keep the fines in the cake structure and the fines will move down the bag on the pulse.

In some cases, bag houses do not have the ability to provide a strong enough regeneration pulse to effectively clean the filter bags. This can be caused by too long of a pulse that drains compressed air from the tank and the off time too short to allow the tank to regain pressure to pulse the next row. The recommended on time for pulse duration is from .05 to .15 seconds. The off time between pulses is site specific due to the compressed airs ability to regain pressure for cleaning. The cleaning pressure should be set at the lowest tank pressure that changes the differential during pulsing. With the settings as mentioned above the collector cannot be cleaned any faster. With the settings cleaning the collector as fast as possible starting and stopping the cleaning based on collector differential change is recommended and the set points should be no greater the 1" apart. It is preferred to maintain the set points at ½" if possible. Differential pressure change of greater than 1" can cause volumetric changes in airflow through the system causing more wear and higher dust loading to the collector. If the cleaning function doesn't change the differential pressure of the bag house the bags will blind due to high pressures and collapse of the dust cake into the bag fabric. Moisture in the gas stream and in the compressed air can accelerate the bag blinding.

Moisture or condensate can form in the blowpipe. This can occur after the pulse for there is no air in the blowpipe and air from the clean air plenum is pulled into the blowpipe to equalize the atmosphere in the pipe to that of the clean air plenum. If the air in the clean air plenum is at temperature and above the dew-point of the gases in the air once pulled into the blowpipe this air sits in the blowpipe extension outside the baghouse at the pulse valve. This extension outside can cool the gas and create acids that will be pulsed into the bags. This will cause cage corrosion and blinding of the fabric.

The two methods for controlling a dust collector are through the use of a circuit board that has the pulse duration at a fixed setting or circuit board that has a photohelic or pressure transducer controlling the cleaning outputs. Older circuit boards do not lend themselves to clean on demand using pressure controls. The recommended circuit boards are ones with memory so when they stop they start at the next row and continue to clean all rows on the manifold before cleaning the same row again. When cleaning on demand it is recommended to locate the pressure taps in the duct prior to the collector and in the duct between the collector and the fan. Using pressure taps on either side of the cell plate or tube sheet can see a needle jump on pulsing that can start and stop the cleaning cycle and not change the differential.

The pulse on demand or clean on demand controls should be set so that pulsing doesn't occur until about 3" of differential is achieved on the bag house and should maintain ½" to 1" of differential change. 3 ½" to 4", or 4" to 4 ½" etc. A dust cake is required on the bag to create efficiency and this usually occurs at or above 3" differential pressure. Pulsing too soon keeps the dust cake from forming and contributes to dusting. Dust that migrates through the bag can also be trapped in the fabric which causes blinding. Not cleaning the bag enough to control the differential pressure will lead to high pressures and this will collapse the dust cake into the fabric and cause blinding.

Pulsing on time can lead to the bags being too clean at times and not clean enough at other times. Too frequent pulsing also consumes considerable amounts of compressed air. According to one manufacturer of valves, pulsing a 1.5" valve every 6 seconds for 0.15 seconds would theoretically consume 67 SCFM of compressed air. The same valve would consume 27 SCFM at 15 seconds intervals and 14 SCFM at 30 seconds intervals. Pulsing on demand with the collector set to clean as fast as possible and cleaning around a maximum of 1" of pressure change is the optimum cleaning cycle for control and will produce longer bag life, compressed air requirement are at the minimum.

(Thanks to Thom Martin of GE Energy for his collaboration in writing up this explanation of baghouse operation.)

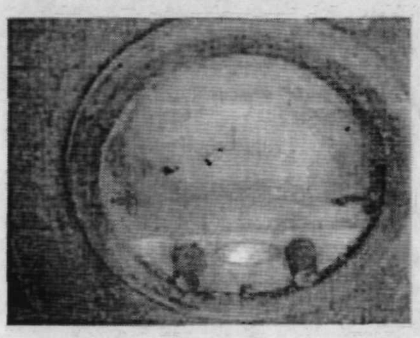
Here are our specific findings.

Solenoid Valve Size – The majority of the pulse jets installed at Ash Grove are 1.5" or larger. A consequence of having these larger pulsejets is that the system demand can increase significantly with slight changes in the settings. This is due to the fact that air consumption increases non-linearly with pulsejet size. For example, one manufacturer estimates that a 2" valve consumes 5.5x the amount of air that a 0.75" valve consumes. Therefore, optimizing the systems means optimizing the controls.

Dust Collector Controls – Since Ash Grove uses primarily POD controls, the first one third of the equation is taken care of and that is pulse interval. The controls for the dust collectors ran the gamut and included remote controls, timer based controls and pulse on demand.

- *Pulse Interval* – Since the units were pulse on demand, the baghouses pulses only when required when set up properly. The only downside to pulsing frequently in a pulse on demand system is that the demand comes on and off quickly. This means that it is possible to have a swing in the load requirement for a short period of time if the dust collectors all through luck or process requirement require air at the same time.

- *Cut in and Cut Out Pressure* - The majority of the settings were a bit lower than is recommended for adequate build up of the dust cake. While there is some savings associated with increasing the settings, the projected savings does not include that as part of the justification. One of the dust collectors was set with the low set point at zero in essence making it a continuous pulse. A picture is to the right of this control. Other settings included the following:



    - 1.5-3"
    - 3.5-3.7"
    - 1.5-2.0"
    - 3.75 -4"
    - .25-.75"
    - 2-2.4"
    - 1-1.5"
- *Pulse Length* - When adequate air quality and air volume exists, pulses lengths of 0.05 to 0.15 are sufficient in most dust collectors to properly pulse the bag. Pulse times varied in the plant and ranged from 0.03 seconds to .35 seconds.
    - A dust collectors in the clinker silo storage area had pulse settings of 0.35 seconds and tests showed it was consuming 28 cfm. This unit apparently only runs 10% of the time so the potential savings was minimal, however, the pulse time should be reduced since there is no cost to do this and the potential always exists for an increase in demand.
    - A dust collector in the clinker storage silo also had a 0.35 second pulse duration. This was not running during the study but should be reset if it does run at all.
    - The finish mill main compressors consumed 48, 90 and 100 cfm with settings of 0.25 and 0.15 for pulse time. Resetting the single unit down to 0.15 seconds duration will reduce air consumption form 48 to 28 Scfm. If it is possible to get effective performance at 0.10 seconds, than the demand drops to 20 Scfm
    - There is a unit on the load out roof that testing showed used 44 Scfm with a duration of 0.35. Resetting this to 0.15 or 0.10 Scfm will save 31 or 37 Scfm.
    - The main coal dust collector pulse times were not known and fairly inaccessible. The plant should review the settings on these units to insure that they too are properly set. Demand at the time of the study for these units was measured at 66 Scfm on one unit and 88 Scfm on the second.

Air Pressure Regulation - Pulsing at higher than required pressures will consume excess air without any gain in pulsing efficiency. All that has to happen is that the bag flexes which is a function of the size of the pulse, not the pressure. Air pressure delivered to the dust collectors was higher than required which results in excess consumption in the form of artificial demand and hence excess cost.

At your plant, pressure ranged from a low of 56 to a high of 125 psig if the gauges are to be believed. Roughly half of the dust collectors were unregulated while those that were regulated supplied air from a low of 56 psig to 125 psig if the gauges are to be trusted.

When there is a sufficient amount of storage installed at the manifold, most dust collectors will operate adequately with a regulated supply pressure of 70 Psig. Once the pulse time is reset down, the ability to pulse at existing pressure should be observed and if the shorter pulse time at the higher pressure lowers the differential, then the differential should be lowered to see whether or not there is an opportunity to further reduce the pressure.

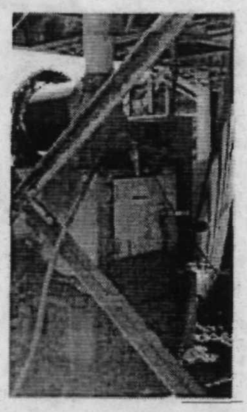
Storage – The majority of dust collectors depended on their manifold for storage and did not have localized storage. The value in a storage tank for a dust collector is that that dust collector is less dependent on air in the manifold and the pipe feeding the dust collector. When there is insufficient storage, the dust collector often requires a higher pressure fed to it to insure there are a sufficient number of molecules to blow the bag. If there isn't, pressure dips during the pulse that results in less cake being removed. The downside of having storage though is that unregulated dust collectors use significantly more volume of air per pulse. The impact the dust collector has on other applications in the area can be improved by the addition of a needle valve that allows air to slowly refill storage so that the required pressure is achieved right before the very next pulse.

Pressure Gauges – Several dust collectors did not have pressure gauges on them and these should be added to all as a maintenance item.

Blown Diaphragms and Leaks - One dust collector was observed on the blending silo roof that had blown the pulsejet. Measurements estimated the demand between 350-400 Scfm which is unusually high and could signify that more than one pulsejet had failed. By the time this report is read, we suspect this will have been long since fixed. Since it was part of the base line of our study and we did not know how long this existed, we did have to treat it as an existing demand and part of the savings stream of the project. Please note that of the 543 Scfm in leaks, 373 Scfm. Between the leaks that were found during th study and this dust collector, we assumed that 343 Scfm could be found and repaired readily.

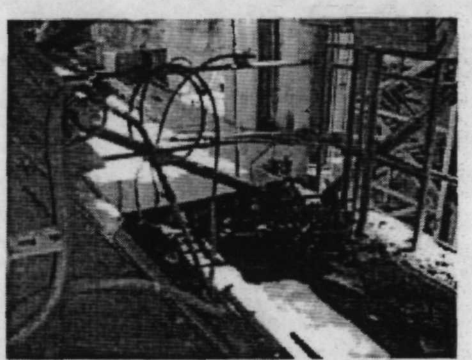
Open Blowing and Aeration

Ash Grove Seattle's operation is one of the most liberal users of compressed air in the plants we have studied. On one hand, the plant has gone to considerable effort to add pulse on demand controls. As an example, the cement dome collector which was off since the dP was 0 during the time of the study would theoretically use 16 Scfm given its current settings. If the unit operates at a 50% load, then the pulse on demand saves 8 Scfm. At the same time, there are air lances blowing 300 Scfm of air in a transfer tower on couple of belt lines that weren't even running.

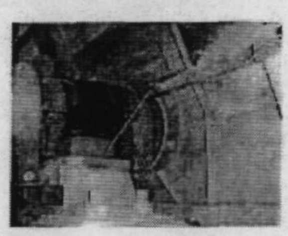


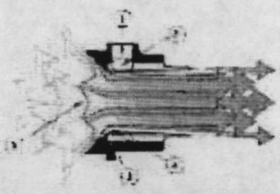
The plant is clearly aware that air is expensive, however, it is clear that the real cost is unknown given the types of decisions that seem to be made at the operator level.

- Open Blowing – Air lances are often required to keep product flowing due to weather and plant configuration. Two lances were observed on a single chute at one of the transfer towers despite the fact that the belt was off. The picture to the right shows an air lance on a three quarter inch line with a valve that was more than 75% open. Assuming the smallest orifice in the line was 0.5" would give a demand in the 100-200Scfm range depending on hose pressure drop. For the sake of the study, we're assuming that the demand is roughly 120Scfm. The picture to the below right shows the unit on the floor below which had similar characteristics except the valve was wide open. Using 240Scfm as the basis, the energy cost alone is estimated at $21000 per year and is more likely in the $28,000 per year when all costs are considered. Discussions with Janike and Johanson suggested not knowing the root cause; they would estimate that a solution for this problem would run in the $5000 to $30,000 range. They indicated that for $5000-$6000, they could review all flow problems in the plant and develop solutions in conjunction with CAC to eliminate the requirement of air lances for this application and others. While we believe that most air lance applications can be solved without the use of compressed air, it is possible that there will be short duration events or other issues that make the use of compressed air the correct financial decision. For these cases, the plant should begin investigating the use of various types of engineered nozzles as manufactured by ITW Vortec or Exair.

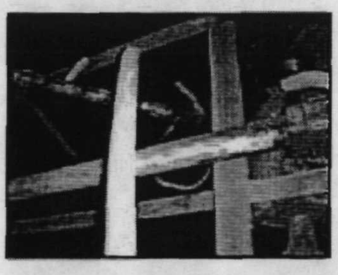


- Bearing Cooling – Bearing cooling was observed in three locations including the coal mill, the VRM classifier and the raw mill fan. The combined demand from these applications is estimated at 135Scfm. There are several ways to reduce the energy associated with this including the use of fans (vane axial or centrifugal) as well or at worst case using an air amplifier and solenoid. An air amplifier uses compressed air to induce air flow up to 25 times as much as the compressed air that is being used. For instance, the bearing cooling such as the 70Scfm used on the VRM could be produced with a 6Scfm air amplifier inducing 73Scfm at the outlet and 219Scfm at a distance of 6".





- Air Slide Assist - Compressed air was found on one of the air slides in the load out area. There were two lines, 0.75" and 0.5" that were 50 and 75% open. Theoretical demand would be 150Scfm and given the pressure drop in the hose, this has been adjusted downward to approximately 80Scfm. This is another application where blower air most likely would be effective and should be evaluated for root cause by

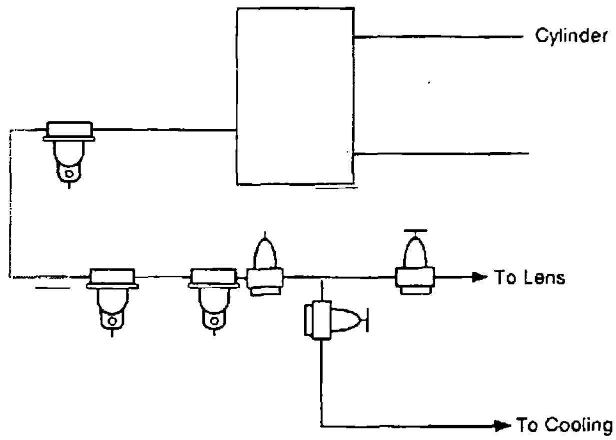


Ash Grove or by an outside firm. From an energy perspective, blower air can cost one third to one tenth the cost of compressed air. In this case, the project financials assume that the savings stream is 40-60Scfm. Potential savings could be even higher.

- Herzog air assist – Compressed air is used for the Herzog machine and the plant is installing a solenoid valve that will actuate for 30 seconds every hour. The estimated demand is 36Scfm and this would reduce the demand to less than 1Scfm and is recommended from an energy perspective.

Air Cannons
Air cannon operation was conservative and no opportunity to reduce demand here was observed.

Cameras
The camera uses compressed air for enclosure cooling, lens clearing and moving the camera in and out. The set up is typically similar to what is shown to the right. In this case, the gauges were unreadable and appeared to have been affected by the heat. It is not known whether or not this is a safety issue, however, the gauges should be replaced. Once the gauges are replaced, the plant should review whether or not the settings are in accordance with the manufacturer's requirement. Typically lens pressure is 5-15 psig and the enclosure pressure requirement is 100 psig. With these types of settings, demand is 50-65Scfm. Reductions in demand here are possible and should be considered in phase two. The two primary ways that demand reduction can be sustained is the conversion of the lens to low pressure blower air or the replacement of the system with a camera that requires no cooling air such as the Pyroviper. This should not be considered an endorsement of the technology as we are unqualified to evaluate cameras; however, there is a cost savings that would be worth approximately $2600 per year.



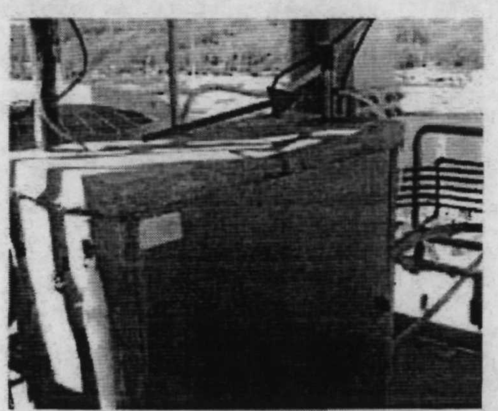
Vibrators
No recommendations are suggested at this time regarding vibrators.

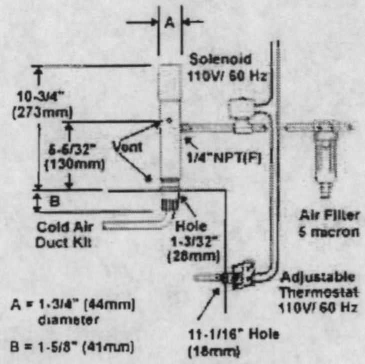
Finish Mill Grinding Aid and Water Atomization
No recommendations are suggested at this time regarding grinding aid or water atomization other than operator education on the criticalness of setting pressure as low as possible. For instance, a nozzle that operates satisfactorily at 15 psig will consume almost twice as much air at 40 psig.

FK Pumps
The FK pumps are operating satisfactorily with the exception of the unit in the load out silo area underneath the ground floor. This unit has a blown solenoid that is consuming significant amounts of air and should be repaired. The pressure on the M250 in the coal mill may be a tad high; however, the potential air savings is minimal.

Cabinet Coolers

There are cabinet coolers on the panel for the water spray system and on a second cabinet on the second floor of the tower. These consume roughly 15Scfm continuous (each) and should be retrofitted with thermostatic controls that turn air on and off according to the ambient temperature. The installed cost for these units is roughly $300. That means that if these turn off the coolers for one quarter of the year, they will have paid for themselves in one year. It is more likely given the climate in Seattle that they will pay for themselves many times over.





Leaks

It is inevitable that during the course of an audit, leaks will be found. While this survey is far from exhaustive, several leaks were found that should be fixed as soon as possible. The plant should also begin a leak detection program.

Location	Eqpt	Application	Size
Old Silos	Separator	Cracked Drain	5
Old Silos- lube area	Separator	Cracked Drain	3
Bldg 331- Roof	Filter	Cracked dP gauge	6
DC between Blend and tower	Vibrator		8
Blending Solenoid 2nd floor	Solenoid	Probable	4
Blending Silo 3rd floor	Solenoid		3
Tower 3rd floor			3
Tower 3rd floor	Air Cannon	Multiple leaks	15
Nitrogen System	Multiple	Valued at 3x	24
Pulverized Coal Feed	Solenoid		6
Finish Mill 2nd floor	Filter	Cracked dP gauge	8
Load Out	Solenoids (4)	Multiple	30
Compressor Room	Auto Drain		50
		Total	165

Several of the leaks are fairly easy to fix including the drains cracked open, the physically cracked dP gauges and the compressor room auto drain. The leaks for the nitrogen system are especially expensive since making those nitrogen leaks took significant amount of air.

In the future, the plant should begin a leak detection that includes both formal and informal leak repairs. While large leaks should be repaired whenever they are found, all leaks should be discovered, tagged and evaluated for repair every three to six months.

This detection program should identify, tag, and quantify leaks in Scfm. The cost to repair the leaks should also be documented during the survey as is possible. Without this information, the plant is not able to make a sound business decision whether to fix a leak or not. Smaller leaks that are not worth repairing should be monitored to insure that they don't grow into large leaks.

Initially, the frequency of a leak detection survey should be a minimum of six months but no more than three moths. The relative return on investment as an outcome of the survey will eventually dictate the frequency of leak detection.

Miscellaneous

Peepers – There is conflicting information on the requirement for the Peepers and we would suggest following up on this application during the detailed engineering phase of the project. Some information suggests that the peepers need 3Scfm at 8" of water column. If pressure is being fed to them at a much higher level, then the demand could be significantly higher.

Pfister Machine and the Coal Hopper – Pfister's and their hoppers often represent an opportunity for further reduction especially when compressed air is used for hopper aeration. Given the criticalness of this application when compared to the potential savings, we are less inclined to improve upon this operation during phase one of a project, however, it should be noted that there is an opportunity. Our best guess is that hopper aeration consumes 40-60 Scfm and could be checked during the detailed engineering phase of the project if Ash Grove wishes to review this application.

Nitrogen Fluffing on the Coal Hopper Bottom - Compressed air costs in the realm of \$100-\$150 per Scfm per year, while nitrogen costs two to three times as much. Operators had indicated that the nitrogen used for the coal hopper bottom could be converted to compressed air if it were dry. While we did not have time to measure this application, it too could be reviewed if the plant is willing to consider converting this.

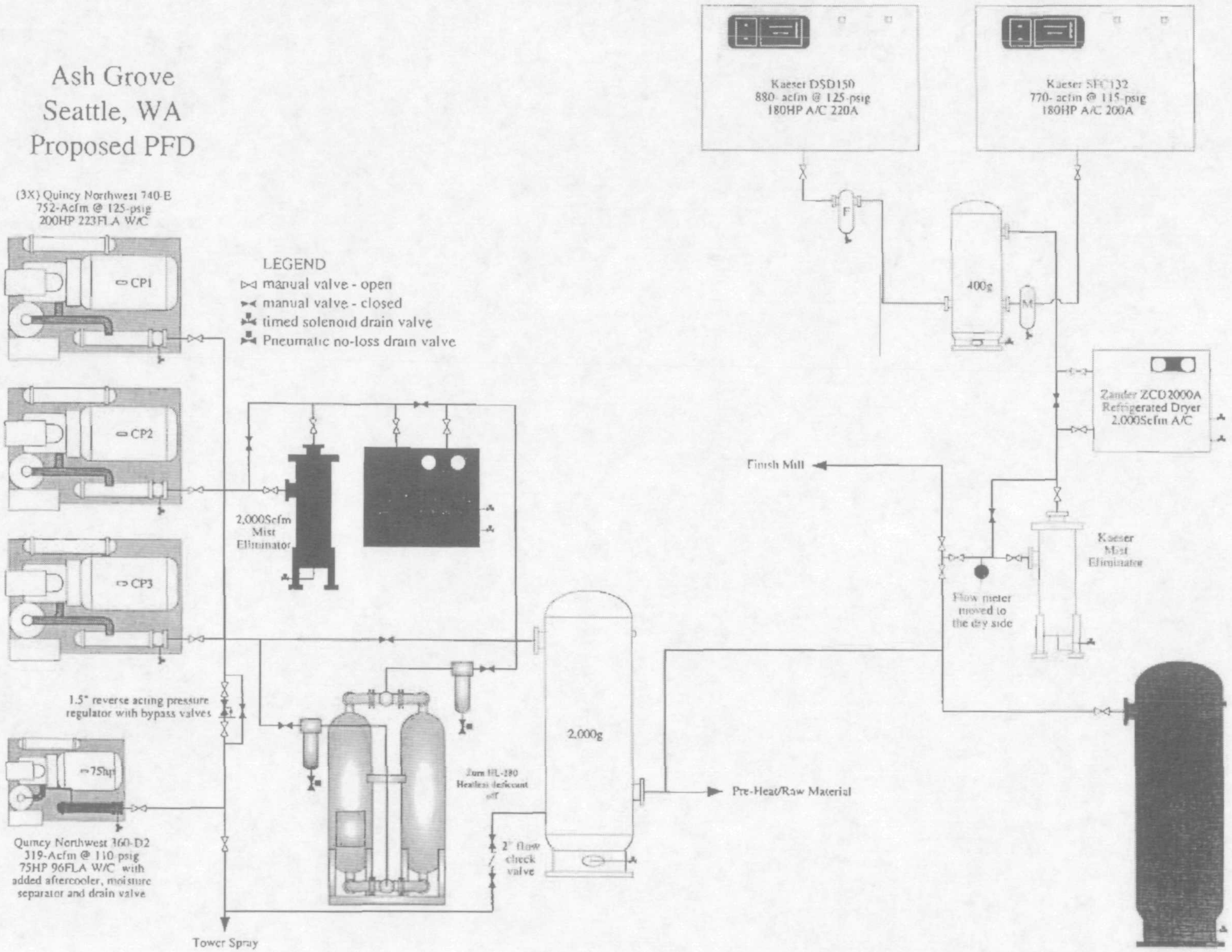
Critical pressure

The area that was thought to be suffering first when plant pressure was low was tower #2. Compressed air is used to feed to two Ø6"X12" cylinders that actuate to position a diverter gate. At certain times, the diverter gate sticks in place, and someone must manually force the gate to one side. Recently, an 11-gallon receiver tank was installed to provide some local volume that could assist the cylinders in their speed of actuation. This did not solve the issue of the gate not moving at times, and due to there not being a drain port on the tank, it was disconnected.

During the audit, the tank, along with a regulator was installed in the line to determine the pressure required to actually move the gate. Pressure was monitored at intervals of 25X/second, and operators sent the signal to move the gate. At an initial pressure of 75psig and a terminating pressure of 32psig, the gate moved, however; there were other times when the initial pressure was 101psig, and the gate would not move. Inspection of the gate and the bin showed that the issues was not air pressure to the cylinders, but an interference fit between the gate and the walls of the bin. It is important that the plant investigate the root cause of issues and not immediately blame compressed air. The maintenance and production team understand that raising plant pressure should be a last resort, and operating the system at a lower pressure is in the best interest of Ash Grove cement. The plant can and should be operated at pressures on or below 85psig.

Section D
Action Plan & Financials

Ash Grove
Seattle, WA
Proposed PFD



Action Item Components

	Item	$/unit	# of units	Equipment	Install
	Supply side actions				
1	Install a reverse acting demand flow control valve on the spray tower compressor	1,000	1	1,000	400
2	Install pneumatic no-loss drain valves on each drain removal point on the supply equipment	600	13	7,800	3,120
3	Install a master signal header and pilot unloader valve so each compressor can operate on the pressure signal located downstream of the cleanup equipment. This should include a 3/4" stainless steel signal pipe taken from the proposed receiver tank to the air compressors.	800	5	4,000	1,600
4	Install 6,000-gallons of storage to be mounted downstream of the dryers in the headerIt should be teed into the line with a service valve as shown.	18,000	1	18,000	7,200
4a	tank painting, pressure gauge kit, safety relief valve kit, drain valve				1,800
5	Install compressor automation that uses Allen-Bradley processor and utilizes rate of pressure change logic.This automation should seamlessly communicate with existing PLC5 platform. This control should be able to force all machines to base load and always trim with the SFC-132	65,000	1	65,000	2,000
6	Install a 2,000 Scfm mist eliminator filter and 2,000 Scfm cycling refrigerant dryer in parallel with the existing desiccant dryer in the Quincy compressor room	30,000	1	30,000	12,000
	Demand side actions				
7	Valve out dust collectors in Quincy compressor room, finish mill.				
8	Perform internal dust collector study detailing controls, settings, storage and pressure. Decrease pulse durations to 0.05 to 0.15, put dP to a 0.5" gap between 3 & 5" water wherever possible. Lower pressure on dust collectors to 70 psig and evaluate adding storage on all units that require a higher pressure than 70 psig. Probable actions include				2,000
9	-Decrease pulse times to 0.15 maximum on separator and mill sweep dust collectors. Evaluate operation at 0.10 second duration				
10	-Repair blown solenoid valve on dust collector on blending silo roof	150	1	150	150
11	-Lower pressure to dust collector on second floor of blending silo from 125 pisg to 80 psig				
12	-Review coal mill dust collector pulse duration and reset to 0.10 or 0.15				
13	Determine root causes requiring compressed air lances in transfer tower and load out air slide assist. Installation cost includes outsourcing of engineering. Capital cost was estimated to range from $5000 to $30,000 by Janike and Johanson based on pictures and descriptions provided to them	20,000	1	20,000	5,000
14	Replace straight bearing cooling of compressed air with air amplifiers, centrifugal or tubeaxial fans.	2,500	1	2,500	
15	Utilize specialty nozzels for all air lance applications in future	300	1	300	

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16	Replace pressure gauges on camera compressed air feed system (Maintenance item)				
17	Repair blown solenoid on load out FK pump	100	1	100	100
18	Add thermostatic control systems to the vortec cooler on the water spray analyzer and the cabinet on the tower second floor	380	1	380	100
19	Repair air leaks including the automatic drain in the compressor room, the air cannons on the 3rd floor of the tower, cracked dp gauge on finish mill second floor and roof of building 331.				500
20	Repair nitrogen leaks in coal mill				100
21	Develop leak detection program that includes tagging and cataloguing of leaks with estimated demand				
22	Purchase Ultrasonic leak detector and perform leak study every three months. Repair leaks as dictated by leak detection program	2,000	1	2,000	
23	Begin a program putting inlet regulators on all dust collectors with pulsejets of 1.5" or greater. Money is allocated to change out ten units as part of the project.				1,500
24	Operator training on the fundamentals of compressed air				1,500
	Freight & Taxes	1.00%	0.00%	1,492	0
	Contingency (10%/20%)	10.00%	20.00%	14,923	7,214
		Sub-Totals		167,645	46,284
		Grand Total		213,929	

	Potential Actions				
	Review use of compressed air on hopper feeding Pfisters				
	Review use of nitrogen on coal hopper bottom				
	Use different platform of Allen Bradley controls (-$20,000 to -40,000)				
	Optional Items				
	Detailed Engineering of Project	20,000			
	Project Management, Assistance or Turnkey	tbd			

Existing annual electrical cost (Compressors)

Rate 0.054 $/KWH

1st shift

	Operating Hours 8760	bhp	kw	hours	kwh	$/kwh	Annual Cost
1	Quincy Norwest - 740-E CP1	154.4	123.3	8760	1,080,406	0.054	$58,342
2	Quincy Norwest - 740-E CP2	146.7	117.2	8760	1,026,539	0.054	$55,433
3	Quincy Norwest - 740-E CP3						
4	Kaeser - DSD150	185.4	148.2	8760	1,298,307	0.054	$70,109
5	Kaeser - SFC132	84.2	67.3	8760	589,632	0.054	$31,840
6	Quincy Norwest - 360-D2 CP2	67.8	55.2	8760	483,188	0.054	$26,092
7	Quincy Norwest - F-30N CP1						
8	Quincy Norwest - F-30N CP2						
				Totals	**4,478,072**		**$241,816**

Existing annual electrical cost (Dryers)

Rate 0.054 $/KWH

1st shift

	Operating Hours 8760	bhp	kw	hours	kwh	$/kwh	Annual Cost
1	ZURN - HL-280	0.0	0.0	8760	0	0.054	$0
2	ZANDER - ZCD2000A	10.0	8.3	8760	72,611	0.054	$3,921
				Totals	**72,611**		**$3,921**

Proposed annual electrical cost (Compressors)

Rate 0.054 $/KWH

1st shift

Operating Hours 8760

		bhp	kw	hours	kwh	$/kwh	Annual Cost
1	Quincy Norwest - 740-E CP1						
2	Quincy Norwest - 740-E CP2						
3	Quincy Norwest - 740-E CP3						
4	Kaeser - DSD150	191.5	153.1	8760	1,341,317	0.054	$72,431
5	Kaeser - SFC132	157.4	125.9	8760	1,102,531	0.054	$59,537
6	Quincy Norwest - 360-D2 CP2	74.1	60.3	8760	528,385	0.054	$28,533
7	Quincy Norwest - F-30N CP1						
8	Quincy Norwest - F-30N CP2						
				Totals	2,972,233		$160,501

Proposed annual electrical cost (Dryers)

Rate 0.054 $/KWH

1st shift

Operating Hours 8760

		bhp	kw	hours	kwh	$/kwh	Annual Cost
1	ZURN - HL-280	0.0	0.0	8760	0	0.054	$0
2	ZANDER - ZCD2000A	10.0	8.3	8760	72,611	0.054	$3,921
new	Cycling Refrig - 2000Scfm	1.8	1.5	8760	13,070	0.054	$706
				Totals	85,681		$4,627

Section E
Attachments

MODEL DATA - FOR COMPRESSED AIR			
1	Manufacturer: Kaeser Compressors, Inc. Date: 6/7/2005		
2	Model Number: DSD 150 - 110 psig [X] Air-cooled [ ] Water-cooled [X] Oil-injected [ ] Oil-free	# of Stages VALUE	1 UNIT
3	Rated Capacity at Full Load Operating Pressure[a,e]	883	acfm[a,e]
4	Full Load Operating Pressure[b]	100	psig[b]
5	Maximum Full Flow Operating Pressure[c]	110	psig[c]
6	Drive Motor Nameplate Rating	150	hp
7	Drive Motor Nameplate Efficiency	95.5	percent
8	Fan Motor Nameplate Rating (if applicable)	3.5	hp
9	Fan Motor Nameplate Efficiency	76	percent
10	Total Package Input Power at Zero Flow[e]	34.3	kW[e]
11	Total Package Power Input at Rated Capacity and Full Load Operating Pressure[e]	143.52	kW[d]
12	Specific Package Input Power at Rated Capacity and Full Load Operating Pressure[f]	16.3	kW/100 cfm[f]

MODEL DATA - FOR COMPRESSED AIR			
1	Manufacturer: Kaeser Compressors, Inc. Date: 8/14/2006		
2	Model Number: SFC 132 - 110 psig / 460V/3ph/60Hz [X] Air-cooled [ ] Water-cooled [X] Oil-injected [ ] Oil-free	# of Stages VALUE	1 UNIT
3	Rated Capacity at Full Load Operating Pressure[a,e]	816	acfm[a,e]
4	Full Load Operating Pressure[b]	100	psig[b]
5	Maximum Full Flow Operating Pressure[c]	110	psig[c]
6	Drive Motor Nameplate Rating	180	hp
7	Drive Motor Nameplate Efficiency	95.5	percent
8	Fan Motor Nameplate Rating (if applicable)	3.5	hp
9	Fan Motor Nameplate Efficiency	76	percent
10	Total Package Input Power at Zero Flow[e]	14.4	kW[e]
11	Total Package Power Input at Rated Capacity and Full Load Operating Pressure[e]	139.86	kW[d]
12	Specific Package Input Power at Rated Capacity and Full Load Operating Pressure[F]	17.1	kW/100 cfm[f]

NOTES

- a Measured at the discharge terminal point of the compressor package in accordance with the CAGI/PNEUROP PN2CPTC2 Test Code (Annex C to ISO 1217); acfm is actual cubic feet per minute at inlet conditions.
- b The operating pressure at which the Capacity (Item 3) and Electrical Consumption (Item 10) were measured for this data sheet.
- c Maximum pressure attainable at full flow, usually the unload pressure setting for load/no load control or the maximum pressure attainable before capacity control begins. May require additional power.
- d Total package input power at other than reported operating points will vary with control strategy.
- e Tolerance is specified in the CAGI/PNEUROP PN2CPTC2 Test Code (Annex C to ISO 1217).
- f, g Tolerance is specified in the CAGI/PNEUROP PN2CPTC2 Test Code (Annex C to ISO 1217) as follows:

Member CAGI

Volume Flow Rate at specified conditions		Volume Flow Rate[f]	Specific Energy Consumption[f]
m³/min	ft³/min	%	%
Below 0.5	Below 15	+/- 7	+/- 8
0.5 to 1.5	15 to 50	+/- 6	+/- 7
1.5 to 15	50 to 500	+/- 5	+/- 6
Above 15	Above 500	+/- 4	+/- 5

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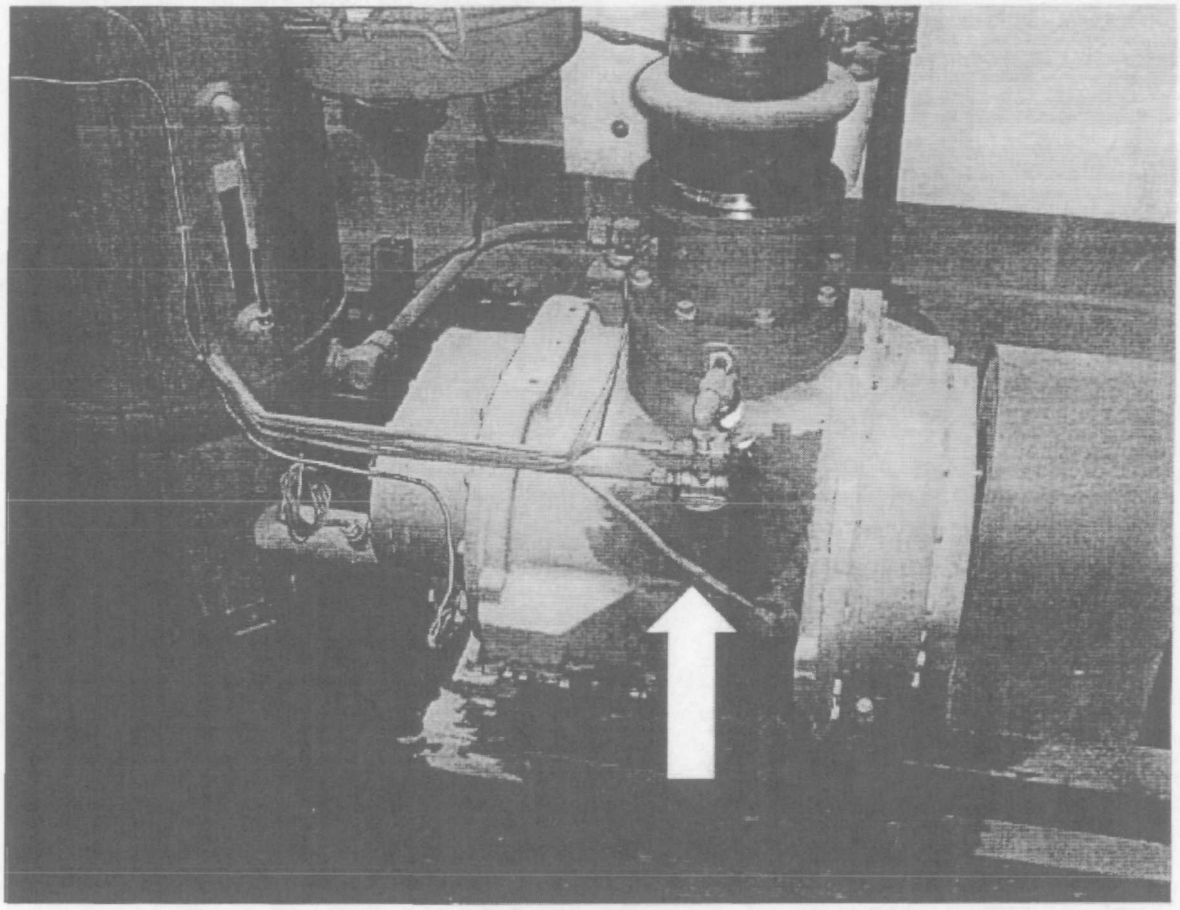
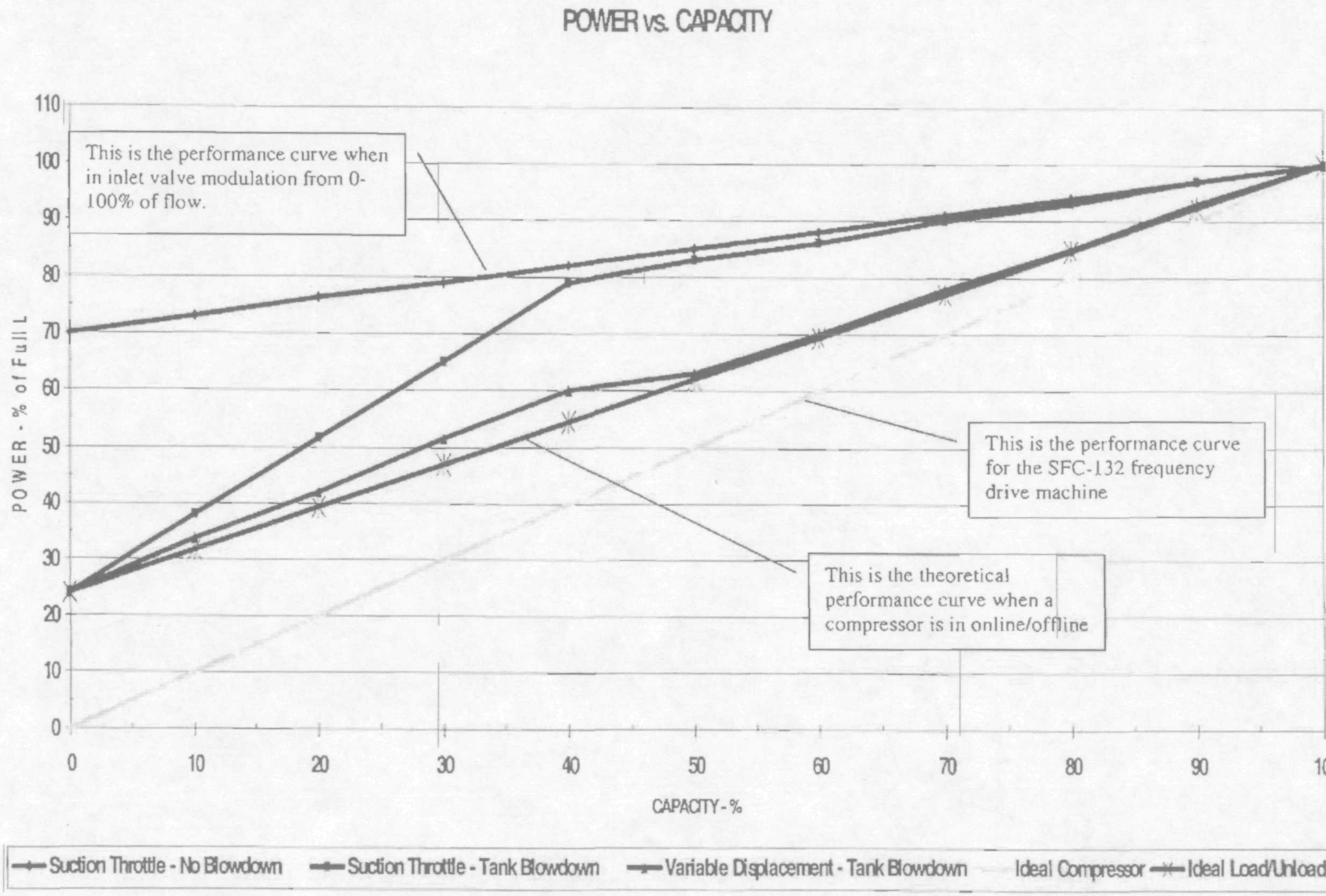


Attachment 1: Subtractive pilot valve - inlet valve control for the Quincy Northwest compressors



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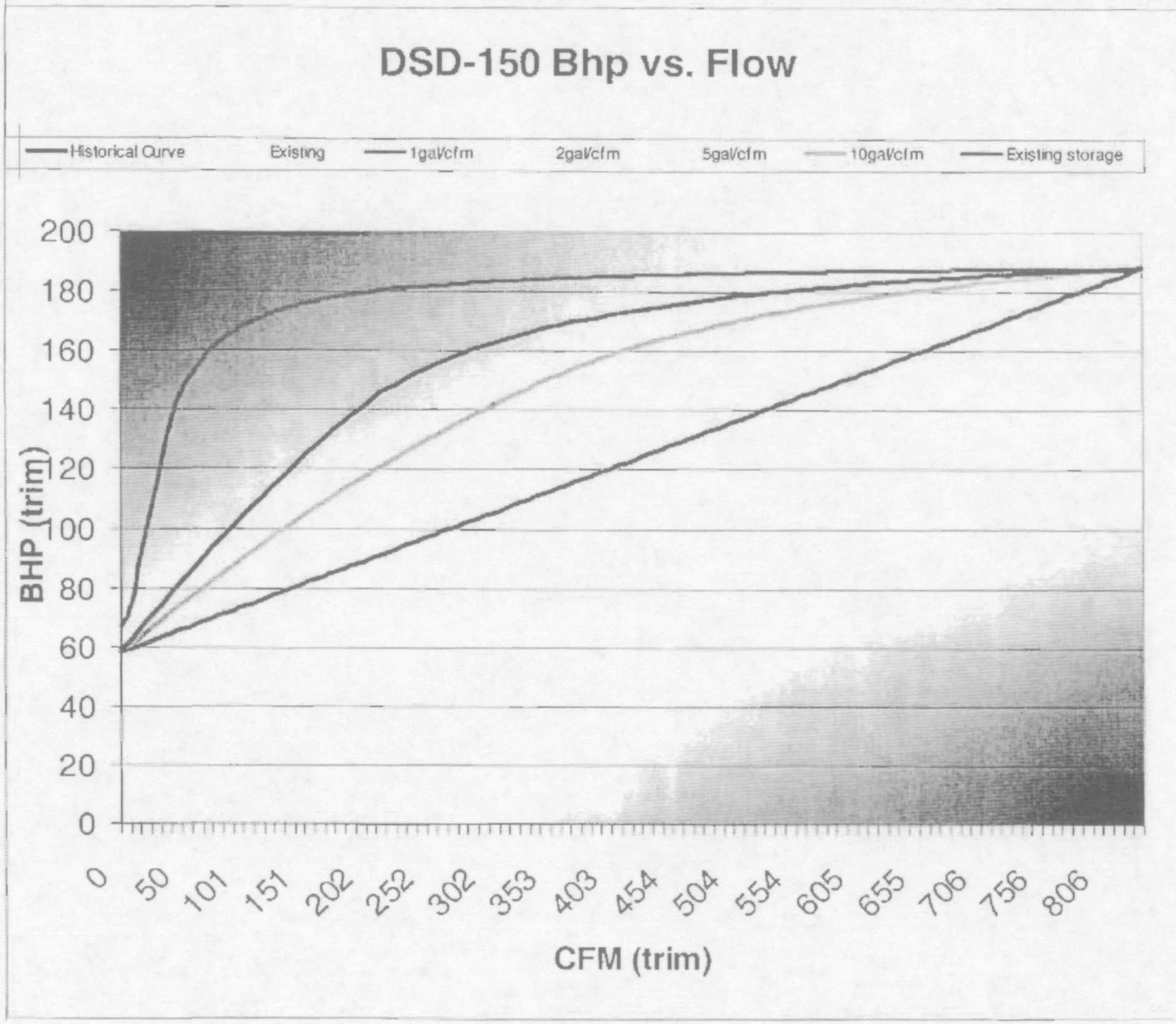
Attachment 2: Specific Power Curves



Attachment 3: Actual specific power curve for the DSD-150

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Moisture Content of Air in Gallons per 1,000 Cubic Feet

	Temperature, ° F									
% RH	35	40	50	60	70	80	90	100	110	120
5	0.0019	0.0024	0.0035	0.0050	0.0071	0.0099	0.0136	0.0186	0.0250	0.0332
10	0.0039	0.0047	0.0069	0.0100	0.0142	0.0198	0.0273	0.0372	0.0501	0.0668
15	0.0058	0.0071	0.0104	0.0150	0.0213	0.0298	0.0411	0.0561	0.0755	0.1007
20	0.0078	0.0095	0.0139	0.0200	0.0284	0.0398	0.0549	0.0750	0.1012	0.1351
25	0.0098	0.0119	0.0174	0.0251	0.0356	0.0498	0.0689	0.0940	0.1270	0.1699
30	0.0117	0.0143	0.0209	0.0301	0.0427	0.0599	0.0828	0.1132	0.1531	0.2051
35	0.0137	0.0166	0.0244	0.0351	0.0499	0.0700	0.0969	0.1325	0.1794	0.2407
40	0.0156	0.0190	0.0279	0.0402	0.0571	0.0801	0.1110	0.1519	0.2060	0.2768
45	0.0176	0.0214	0.0314	0.0453	0.0644	0.0903	0.1251	0.1715	0.2328	0.3133
50	0.0195	0.0238	0.0349	0.0503	0.0716	0.1005	0.1394	0.1912	0.2598	0.3502
55	0.0215	0.0262	0.0384	0.0554	0.0789	0.1107	0.1537	0.2110	0.2871	0.3876
60	0.0235	0.0286	0.0419	0.0605	0.0861	0.1210	0.1681	0.2310	0.3146	0.4254
65	0.0254	0.0310	0.0454	0.0656	0.0934	0.1313	0.1825	0.2511	0.3424	0.4637
70	0.0274	0.0334	0.0490	0.0707	0.1007	0.1417	0.1970	0.2713	0.3705	0.5025
75	0.0294	0.0358	0.0525	0.0758	0.1081	0.1521	0.2116	0.2917	0.3988	0.5418
80	0.0313	0.0382	0.0560	0.0810	0.1154	0.1625	0.2263	0.3122	0.4273	0.5816
85	0.0333	0.0406	0.0596	0.0861	0.1228	0.1730	0.2410	0.3328	0.4562	0.6219
90	0.0353	0.0430	0.0631	0.0913	0.1302	0.1835	0.2559	0.3536	0.4853	0.6627
95	0.0372	0.0454	0.0666	0.0964	0.1376	0.1940	0.2707	0.3745	0.5147	0.7041
100	0.0392	0.0478	0.0702	0.1016	0.1450	0.2046	0.2857	0.3956	0.5443	0.7460

Moisture Content of Saturated Air in Gallons per 1,000 Standard Cubic Feet

	Temperature, ° F									
psig	35	40	50	60	70	80	90	100	110	120
0	0.0392	0.0479	0.0702	0.1016	0.1450	0.2046	0.2857	0.3956	0.5443	0.7460
10	0.2333	0.0283	0.0416	0.0600	0.0854	0.1200	0.1667	0.2290	0.3119	0.4217
20	0.0165	0.0201	0.0295	0.0426	0.0605	0.0849	0.1176	0.1612	0.2186	0.2939
30	0.0128	0.0156	0.0229	0.0330	0.0469	0.0657	0.0909	0.1243	0.1682	0.2256
40	0.0105	0.0128	0.0187	0.0269	0.0383	0.0536	0.0741	0.1012	0.1367	0.1830
50	0.0089	0.0108	0.0158	0.0228	0.0323	0.0452	0.0625	0.0853	0.1152	0.1540
60	0.0077	0.0093	0.0137	0.0197	0.0280	0.0391	0.0540	0.0737	0.0995	0.1329
70	0.0068	0.0082	0.0121	0.0174	0.0246	0.0345	0.0476	0.0649	0.0876	0.1169
80	0.0060	0.0074	0.0108	0.0155	0.0220	0.0308	0.0425	0.0580	0.0782	0.1043
90	0.0055	0.0067	0.0098	0.0140	0.0199	0.0279	0.0385	0.0524	0.0706	0.0942
100	0.0050	0.0061	0.0089	0.0128	0.0182	0.0254	0.0351	0.0478	0.0644	0.0858
110	0.0046	0.0056	0.0082	0.0118	0.0167	0.0234	0.0323	0.0439	0.0592	0.0789
120	0.0043	0.0052	0.0076	0.0109	0.0155	0.0216	0.0298	0.0407	0.0548	0.0729
130	0.0040	0.0048	0.0071	0.0102	0.0144	0.0201	0.0278	0.0378	0.0509	0.0678
140	0.0037	0.0045	0.0066	0.0095	0.0135	0.0188	0.0260	0.0354	0.0476	0.0634
150	0.0035	0.0042	0.0062	0.0089	0.0126	0.0177	0.0244	0.0332	0.0447	0.0595
160	0.0033	0.0040	0.0058	0.0084	0.0119	0.0167	0.0230	0.0313	0.0421	0.0561
170	0.0031	0.0038	0.0055	0.0080	0.0113	0.0158	0.0217	0.0296	0.0398	0.0530
180	0.0029	0.0036	0.0052	0.0073	0.0107	0.0149	0.0206	0.0281	0.0378	0.0503
190	0.0028	0.0034	0.0050	0.0072	0.0102	0.0142	0.0196	0.0267	0.0359	0.0478
200	0.0027	0.0032	0.0048	0.0068	0.0097	0.0136	0.0187	0.0254	0.0342	0.0455

A "positive displacement" compressor is normally rated in ACFM (Actual Cubic Feet per Minute). This is the amount of air, taken from atmospheric conditions, which the unit will deliver at its discharge. Within a broad range, changes in inlet air temperature, pressure, and humidity do not change the ACFM rating of either the reciprocating or rotary screw compressors.

In an effort to obtain an "apples to apples" comparison of various compressors, many firms specify their capacity requirements in SCFM (Standard Cubic Feet per Minute). This sometimes causes much confusion because many people do not full understand how to convert ACFM to SCFM.

$$SCFM = ACFM \times \left[\frac{P_b - (RH_a * PV_a)}{P_s - (RH_s * PV_s)}\right] \times \left[\frac{T_s}{T_a}\right] \times \left[\frac{P_a}{P_b}\right]$$

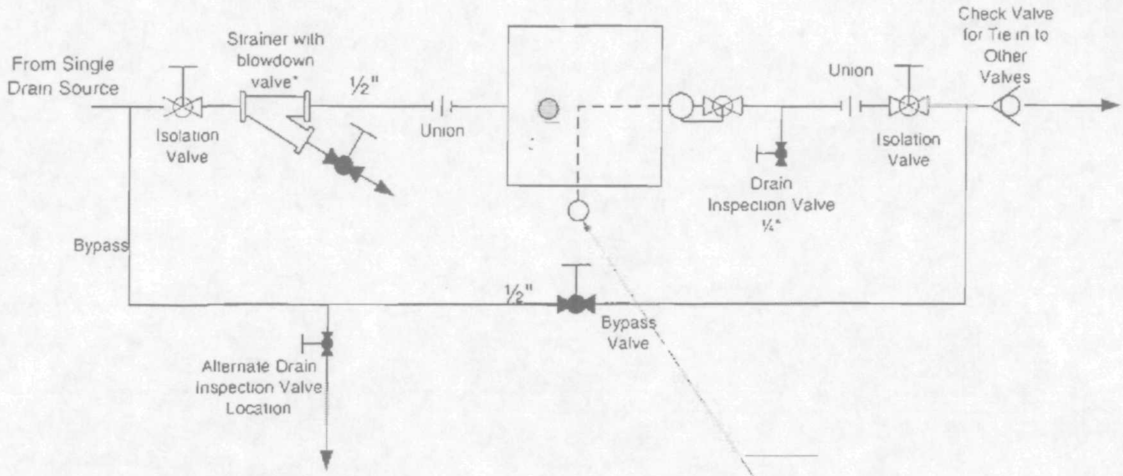
Where

P_s	=	Standard pressure at atmosphere
P_b	=	Actual atmospheric pressure (psia)
P_a	=	Actual inlet pressure (psia) (includes inlet filter and pipe losses)
RH_s	=	Standard relative humidity
Rh_a	=	Actual relative humidity
PV_s	=	Saturated vapor pressure of water at standard temperature (psi)
PV_a	=	Saturated vapor pressure of water at actual temperature (psi)
T_s	=	Standard temperature (°R) note: °R=°F+460
Ta	=	Actual temperature (°R)

Pressure of Water Vapor at Saturation					
Temperature (°F)	Pressure (psia)	Temperature (°F)	Pressure (psia)	Temperature (°F)	Pressure (psia)
32	0.08854	60	0.2563	86	0.6152
34	0.09603	62	0.2751	88	0.6556
36	0.10401	64	0.2951	90	0.6982
38	0.11256	66	0.3164	92	0.7432
40	0.1217	68	0.339	94	0.7906
42	0.1315	70	0.3631	95	0.8153
44	0.14199	72	0.3886	96	0.8407
46	0.15323	74	0.4156	98	0.8935
48	0.16525	76	0.4443	100	0.9492
50	0.17811	78	0.4747	102	1.0078
52	0.19182	80	0.5069	104	1.0695
54	0.20642	82	0.541	106	1.1345
56	0.222	84	0.5771	108	1.2029
58	0.2386	85	0.5961	110	1.2748

Installation of No Loss Drains

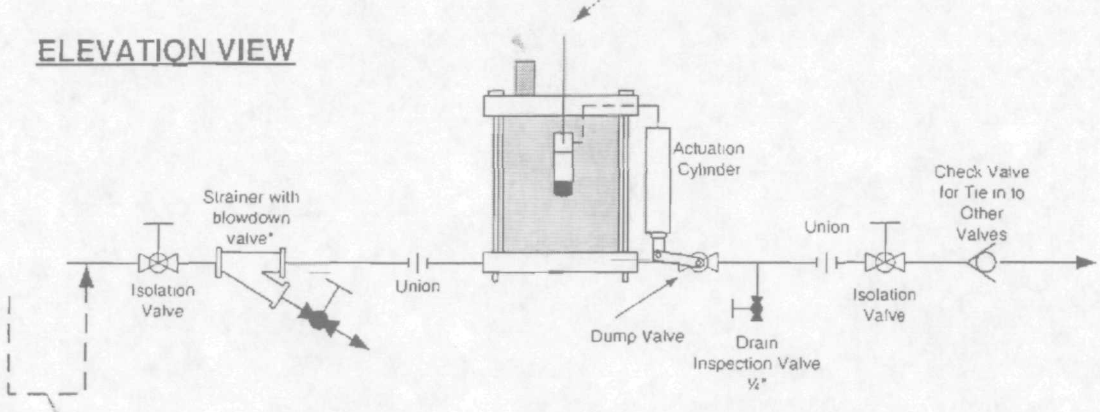
PLAN VIEW



Whisper Valve Vent
Use only the needle valve. Do not install the down stream vent line, which can convey wet air down stream of condensate trap on dryer or water separator. Uses minimal cfm.

Control Air
Actuation air from clean dry source, min 50 psig to pneumatic solenoid controller

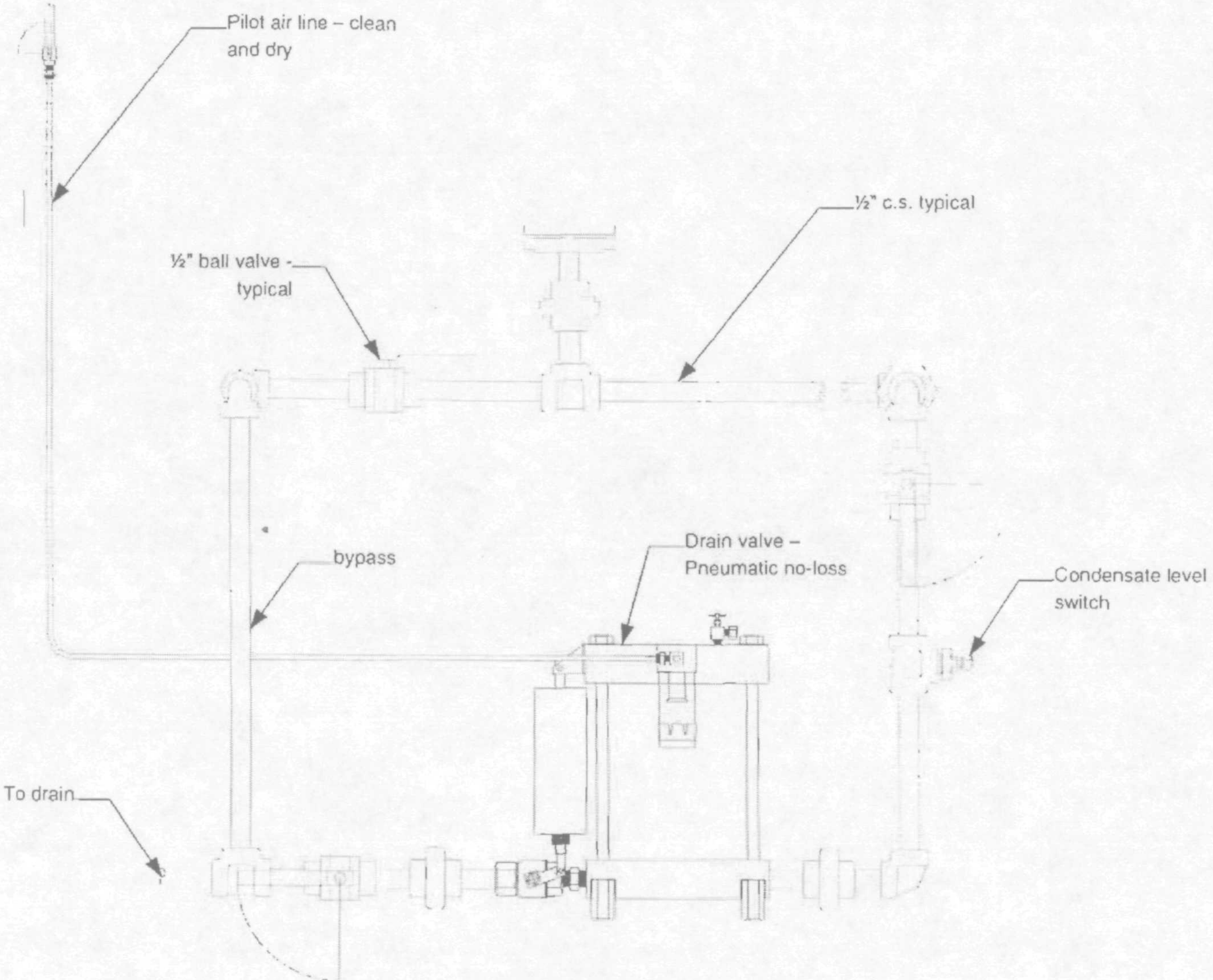
ELEVATION VIEW



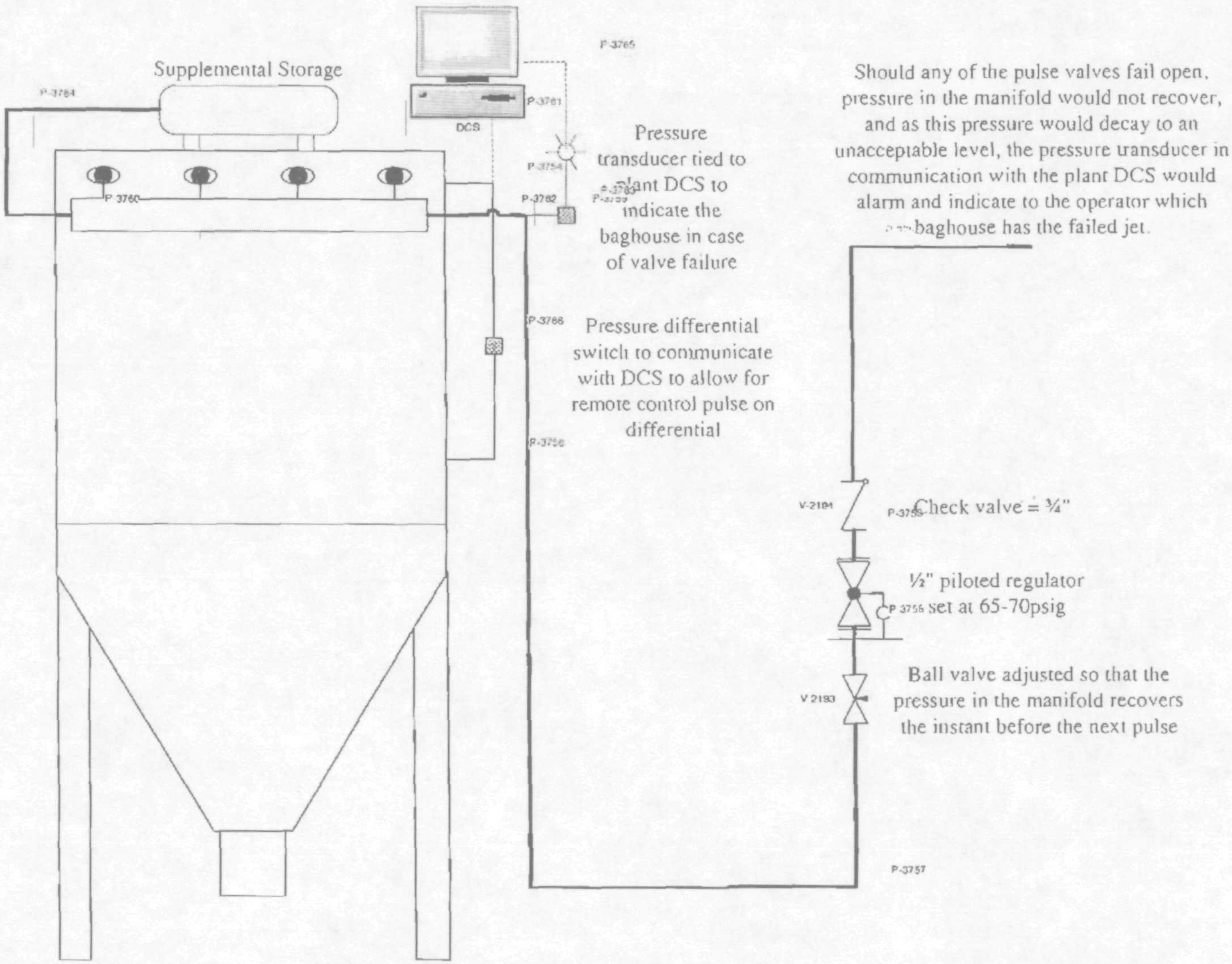
Inlet piping must be at or above the inlet to the drain. Piping below the inlet can create a trap for liquid accumulation which can slug the drain creating a maintenance problem.

*Strainers are recommended for the DrainAll units whereas they are not required for Precision Pneumatic units.

For Ingersoll-Rand centrifugals only.
The intercoolers may have two drain points, which can be connected to a common drain valve, one for each stage. Each line from the intercooler is to have an isolation valve.



Drain valve installation



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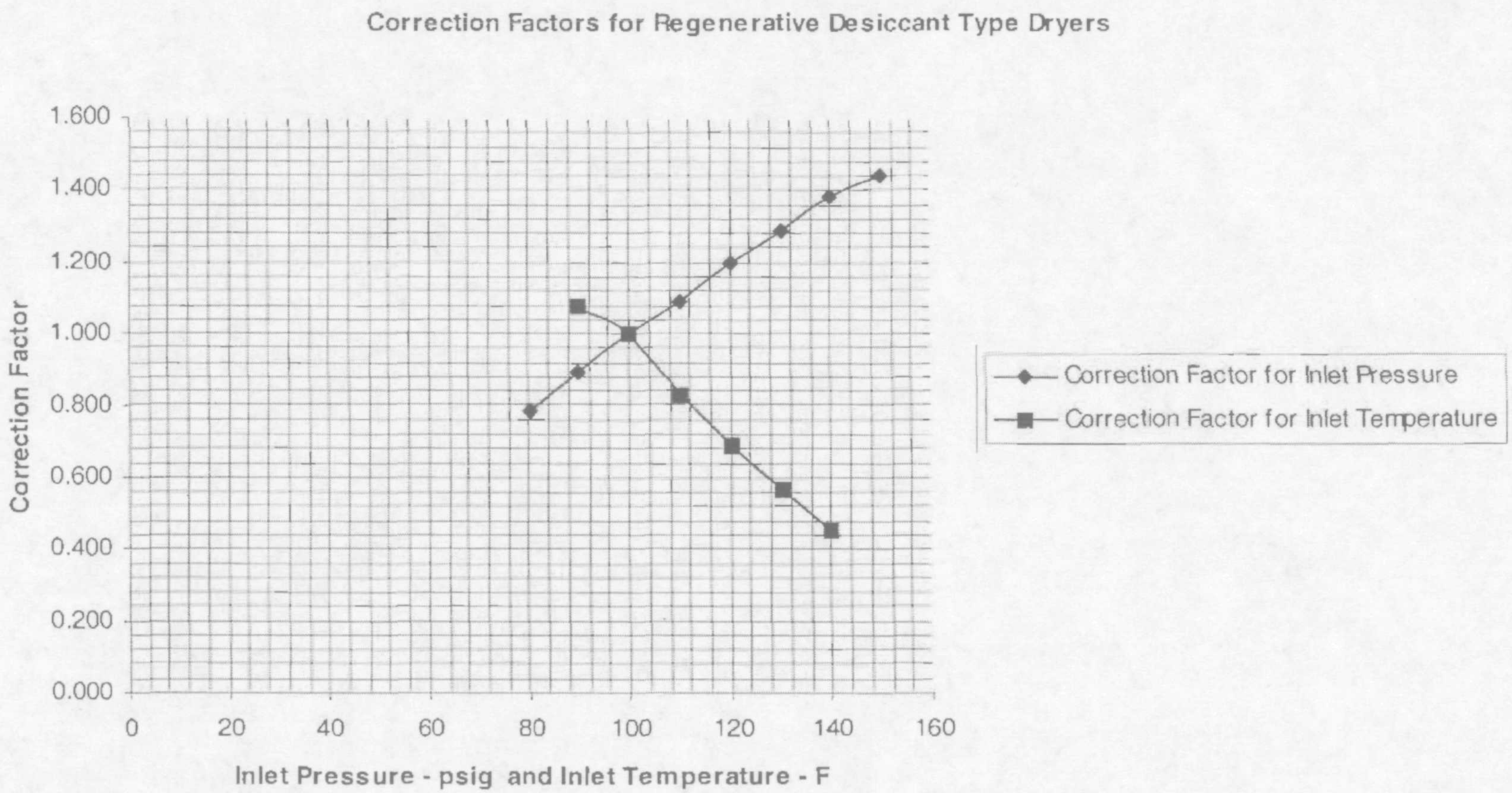
Proper installation of local storage and isolation to a baghouse



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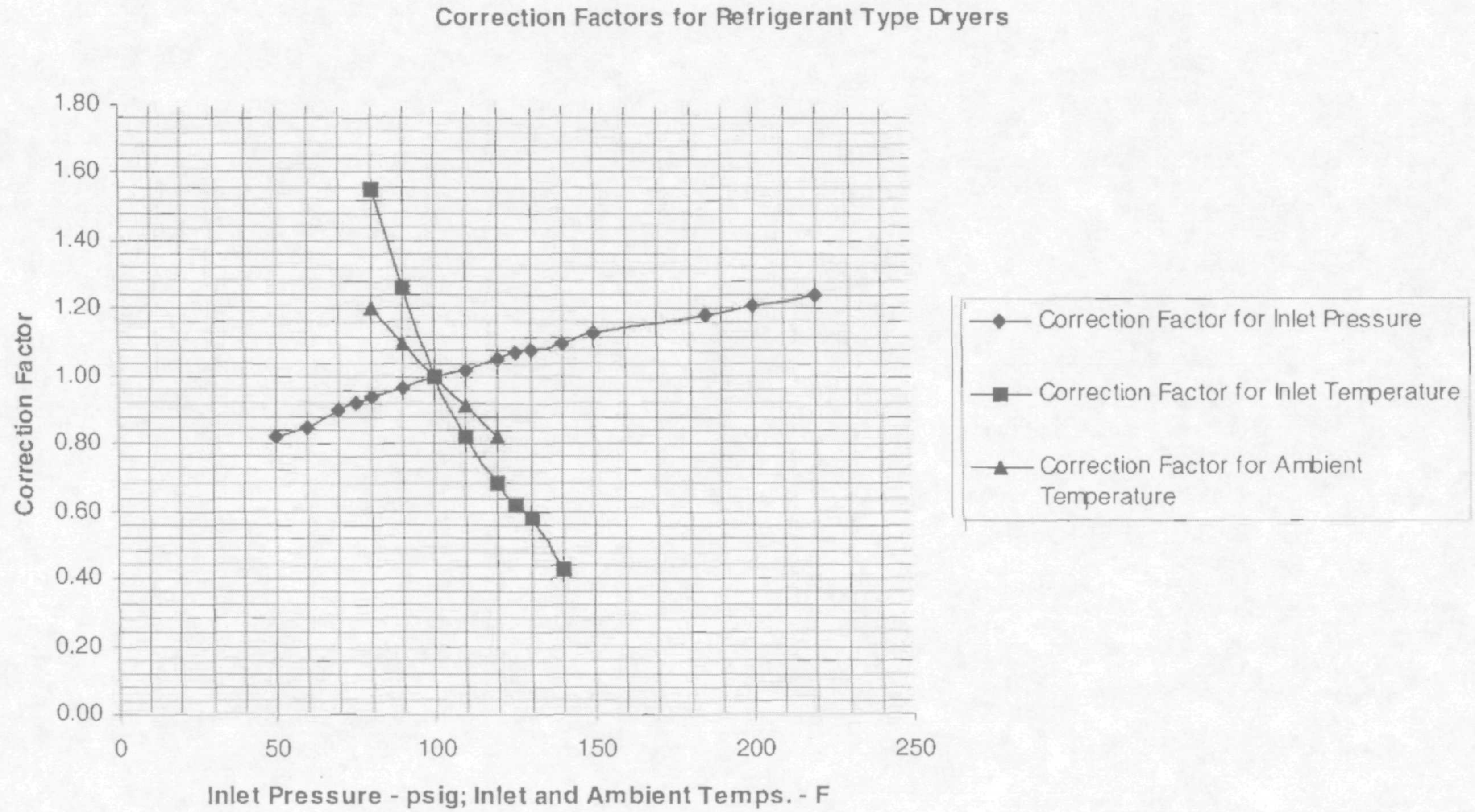

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26.4 KV electricity ("power") is delivered to the plant by the local supplier Seattle City Light. The power enters the plant through one of two main substations. These two substations will be referred to as "Old" and "New". Below will be a description in bullet form of how the power routes from each of the substations through transformers and motor control centers to the individual motors.

"Old" Substation

- Power from the old substation is stepped down to 4,160 volts and runs to the Finish Mill Building Motor control center. From the finish mill motor control center the power is run to one of six transformers that service various electrical areas of the plant. For descriptive purposes only the transformers are numbered 1 through 6.
    - From transformer #1 the power is further stepped down in voltage and routed to the individual motor breakers, or buckets, and then routed out to the individual motors for the finish milling process.
    - From transformer #2 the power is further stepped down in voltage and routed to the main office, maintenance shop, and cooling water systems.
    - From transformer #3 (also known as the Group II transformer) the power is further stepped down in voltage and routed to the individual motor breakers, or buckets, and then routed out to the individual motors for the shipping process.
    - From transformer #4 (also known as the coal transformer) the power is further stepped down in voltage and routed to the individual motor breakers, or buckets, and then routed out to the individual motors of the coal grinding process.
    - From transformer #5 (also known as the dome transformer) the power is further stepped down in voltage and routed to the individual motor breakers, or buckets, and then routed out to the individual motors of the dome storage system.
    - From transformer #6 (also known as the clinker transformer) the power is further stepped down in voltage and routed to the individual motor breakers, or buckets, and then routed out to the individual motors of the clinker storage and barge loading systems.

"New" Substation

- Power from the new substation is stepped down to 4,160 volts and routed to three different motor control center transformers and two system breakers, or buckets.
    - From the #158 motor control center's transformer the power is further stepped down in voltage and routed to the individual motor breakers, or buckets, and then routed out to the individual motors of the barge unloading and raw material reclaim system.
    - From the #448 Burner Building motor control center's transformer the power is further stepped down in voltage and routed to the individual

motor breakers, or buckets, and then routed out to the individual motors of the kiln system.

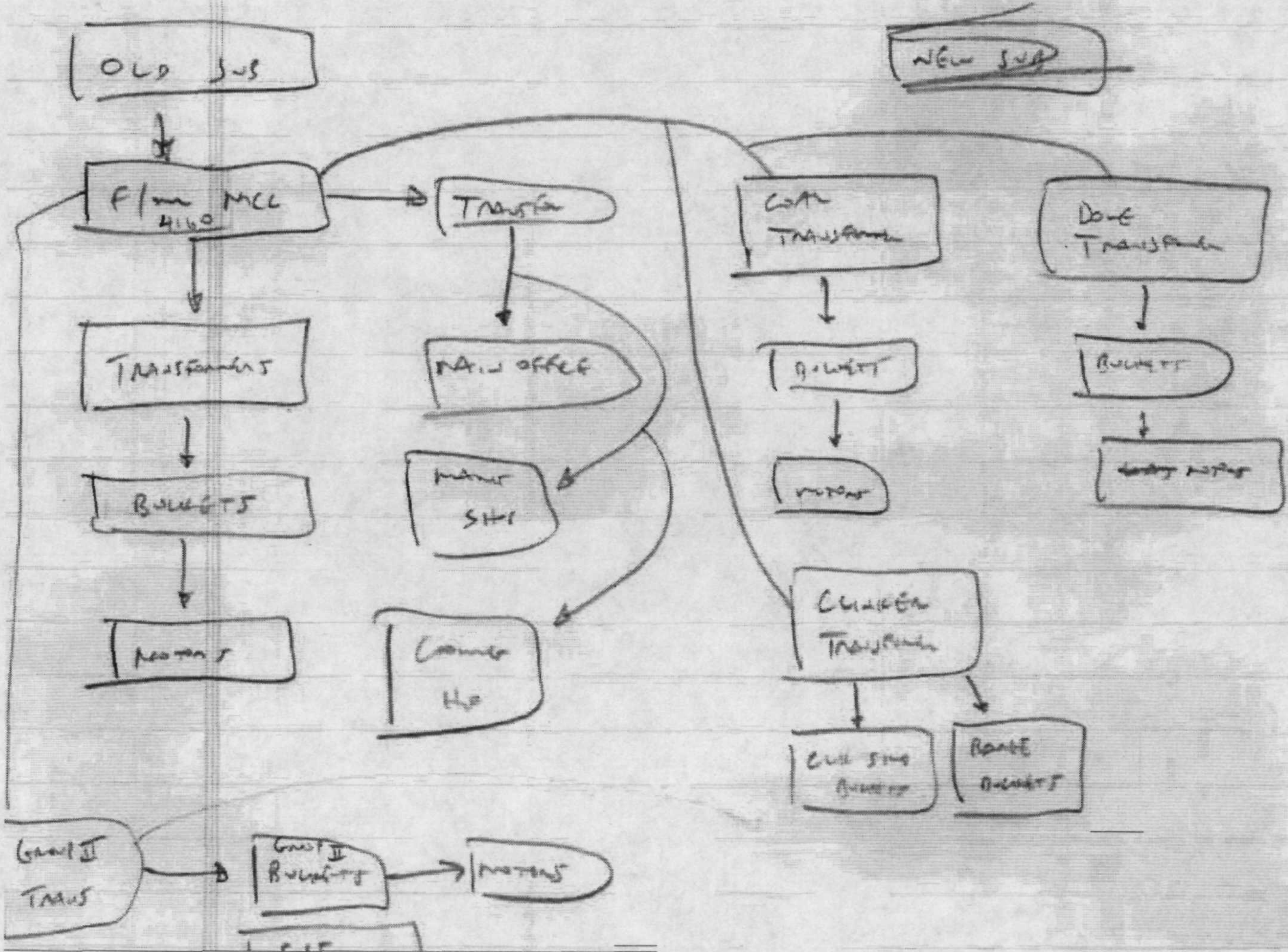
- From the #348 motor control center's transformer the power is further stepped down in voltage and routed to the individual motor breakers, or buckets, and then routed to the individual motors of the raw mill, kiln, and baghouse systems. 4,160 volt power from this motor control center is also fed directly to the raw mill, raw mill fan, kiln induced draft fan, baghouse fan, and kiln drive motors.
- Power from the substation is routed directly to the cyclonaire (barge loading) system.
- Power from the substation is routed directly to the stiff leg crane motor. The stiff leg crane is part of the barge unloading system.

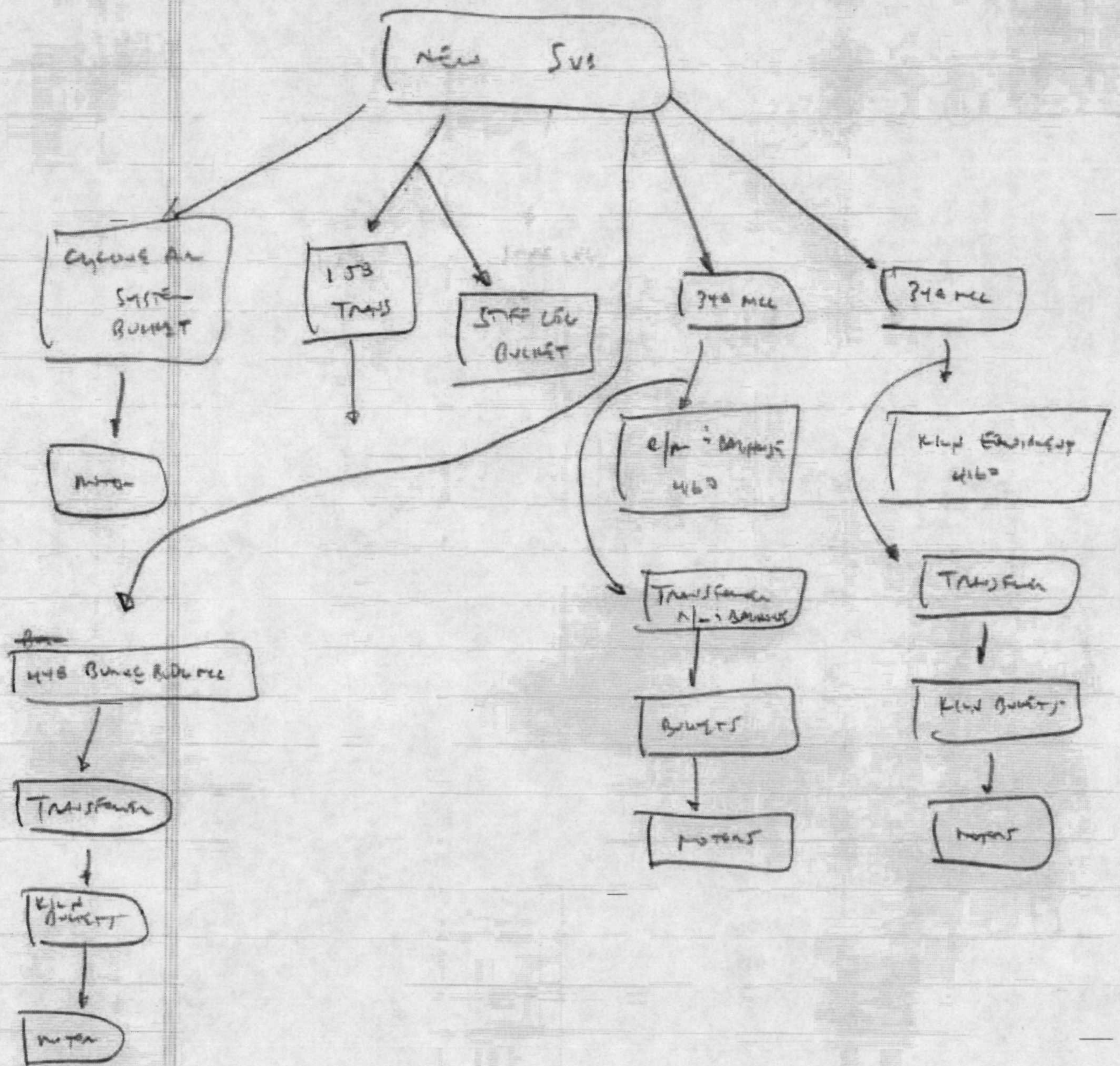
When incoming power is lost from Seattle City Light, a diesel generator is started in order to supply power to critical systems. The power from the generator is routed to the two main substations and then through automatic transfer switches it is diverted to specific motors in the kiln, coal mill, burning, and finish milling systems.

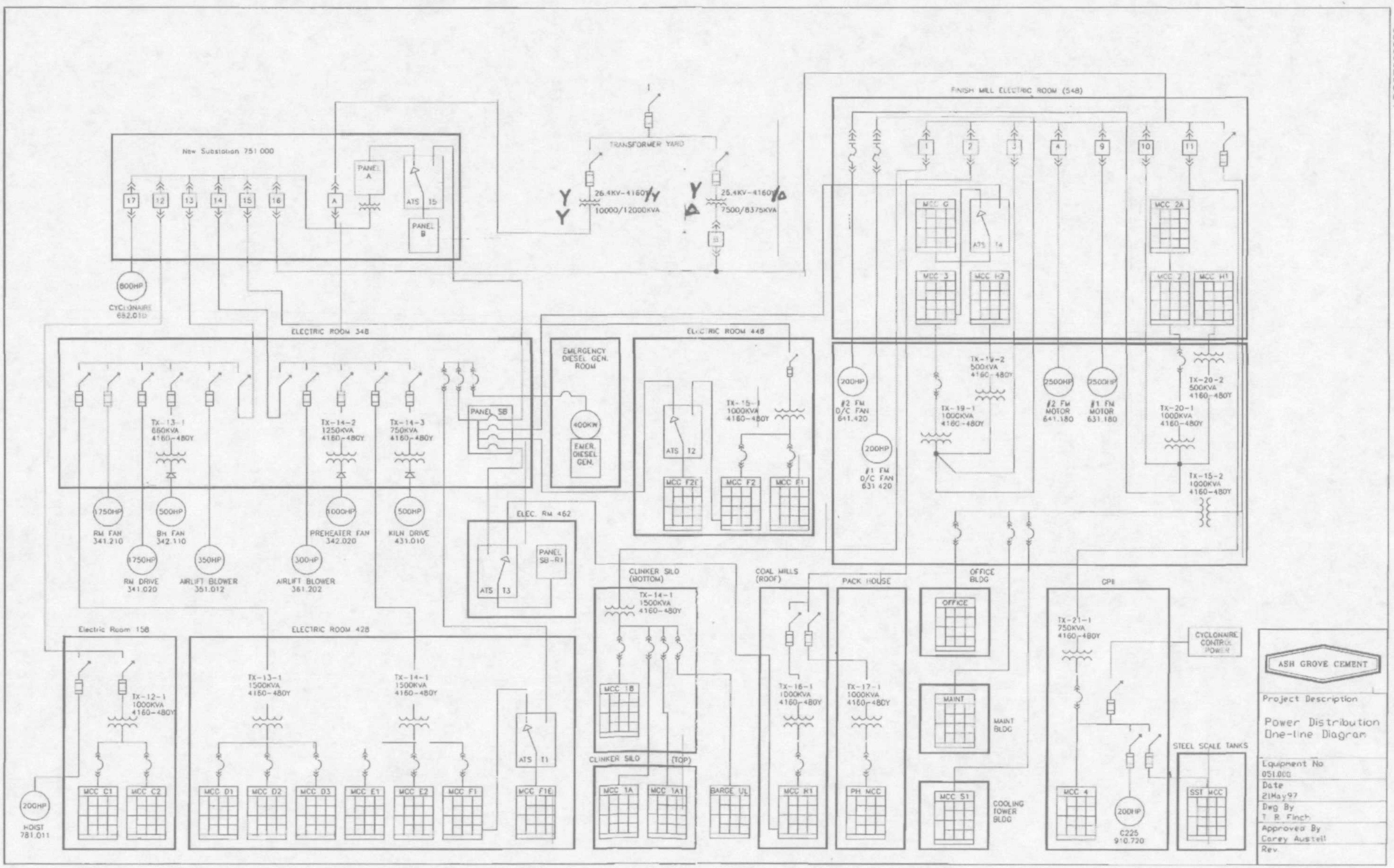
TWO
4160 V POWER SCL TO MAIN SUB

ATS ARE BACK FED FROM THE ~~DIES~~ DIESEL GEN.

- 423 MCC KILN
- 462 MCC COAL MILL
- 442 MCC BURNER BLDG.
- NEW SUB
- 543 MCC FINISH MILL




STIFF LEG BUCKET
TRANSFER
KILN BUCKETS
MOTOR
BUCKETS
MOTORS



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